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Meacham et al.

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(54) **ANISOTROPIC BEARING SUPPORTS FOR TURBOCHARGERS**

(75) Inventors: **Walter Lee Meacham**, Phoenix, AZ (US); **Kostandin Gjika**, Epinal (FR); **Mohsiul Alam**, Chandler, AZ (US); **Gerald D. LaRue**, Torrance, CA (US)

(73) Assignee: **Honeywell International Inc.**, Morristown, NJ (US)

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Related U.S. Application Data

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F04B 35/00 (2006.01)
F16C 27/00 (2006.01)

(52) **U.S. Cl.**
USPC **417/407**; 384/119; 384/215; 384/901

(58) **Field of Classification Search**
USPC 417/407; 384/99, 119, 215, 901
See application file for complete search history.

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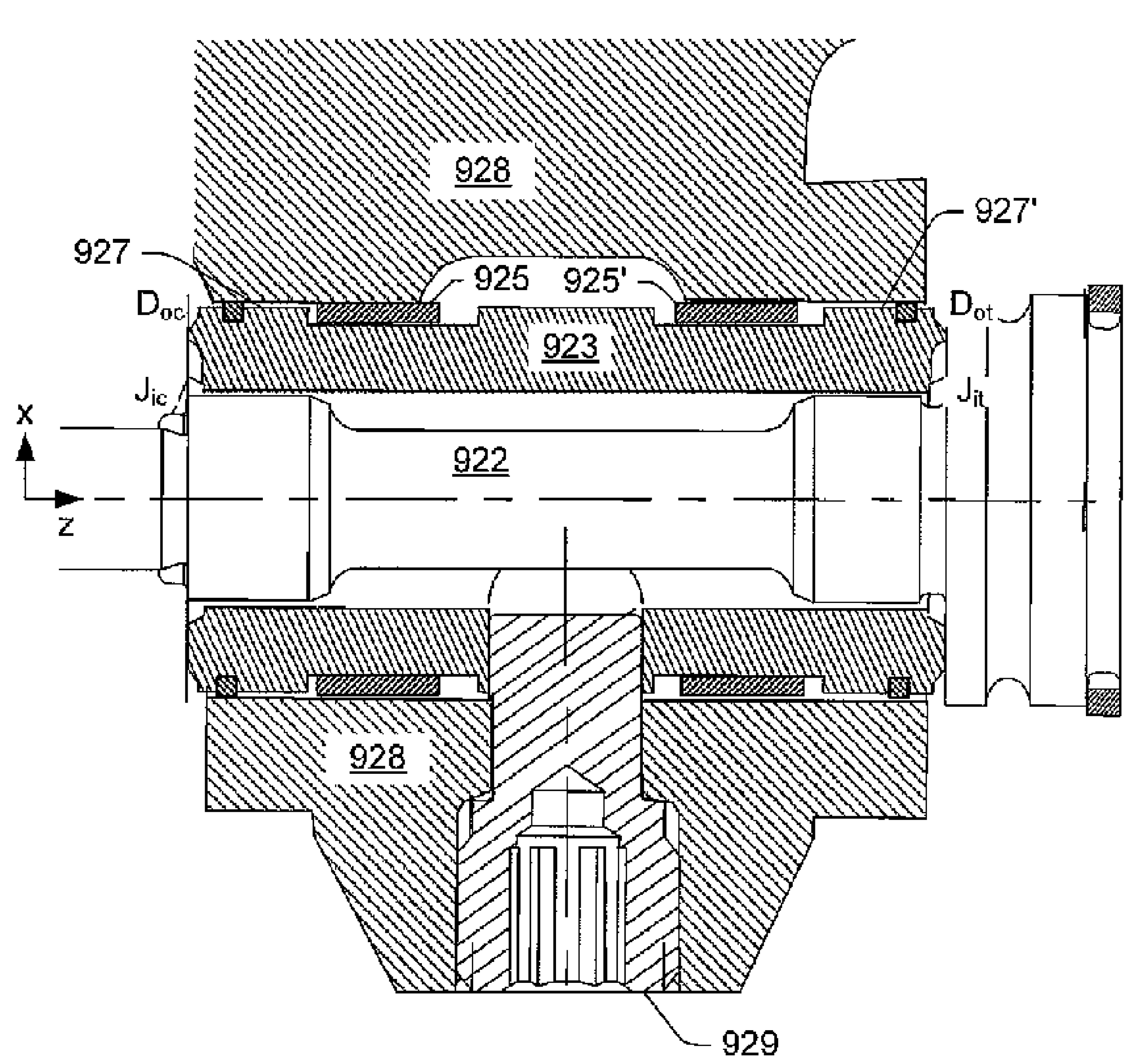
Primary Examiner — Charles Freay
(74) *Attorney, Agent, or Firm* — Brian J. Pangrle

(57) **ABSTRACT**

An exemplary anisotropic member supports a semi-floating bearing in a bore and can reduce non-synchronous vibration of the bearing in the bore. An anisotropic member can include an annular body configured to receive a semi-floating bearing and to space the bearing a distance from a bore surface and can be configured to impart anisotropic stiffness and damping terms to the semi-floating bearing when positioned in the bore. Such a member is suitable for use in a rotating assembly for a turbocharger where the bearing may be a semi-floating bearing. Various exemplary members, bearings, housings, assemblies, etc., are disclosed.

22 Claims, 19 Drawing Sheets

Exemplary Semi-Floating Assembly with Rings 900



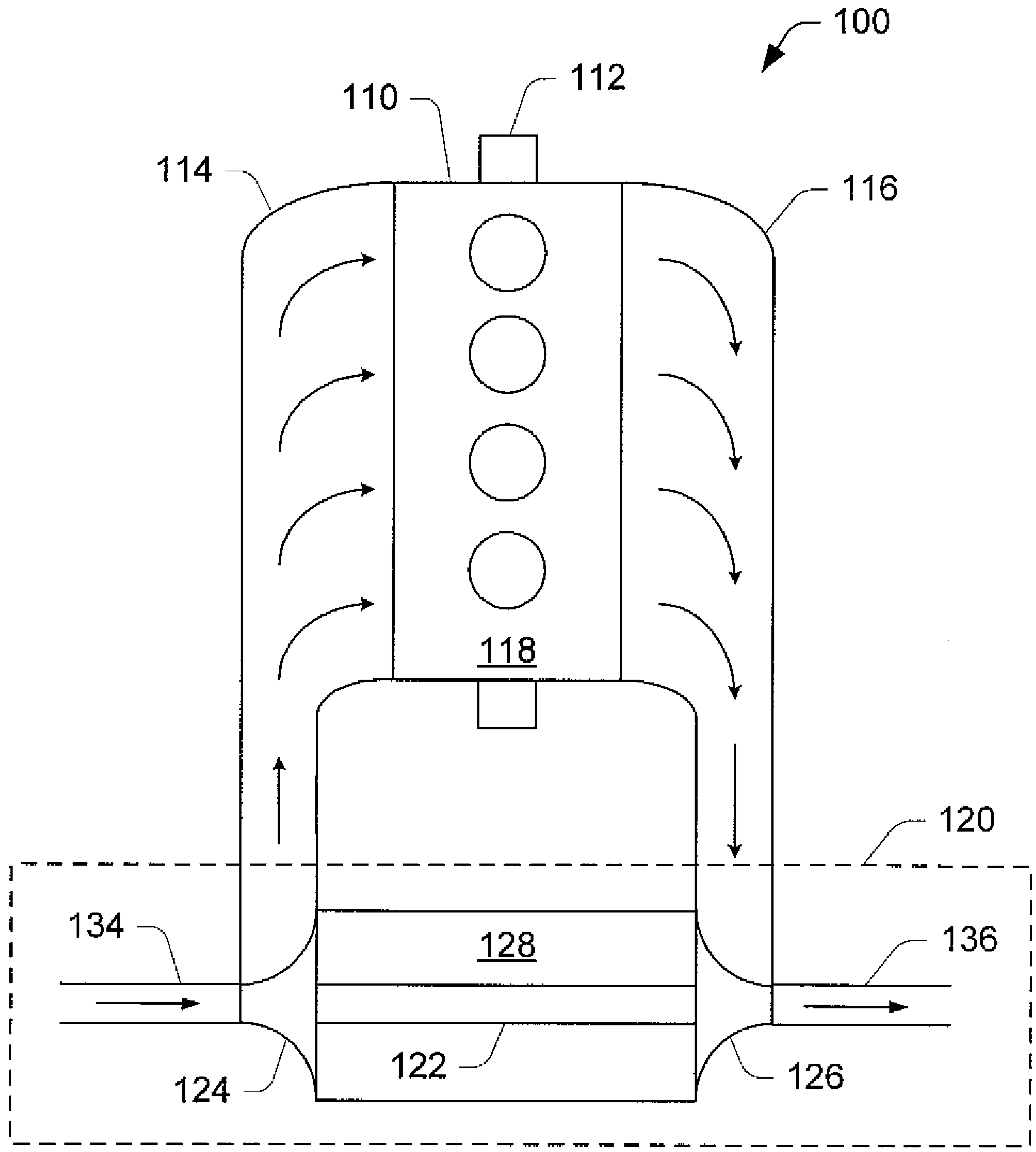


Fig. 1
(Prior Art)

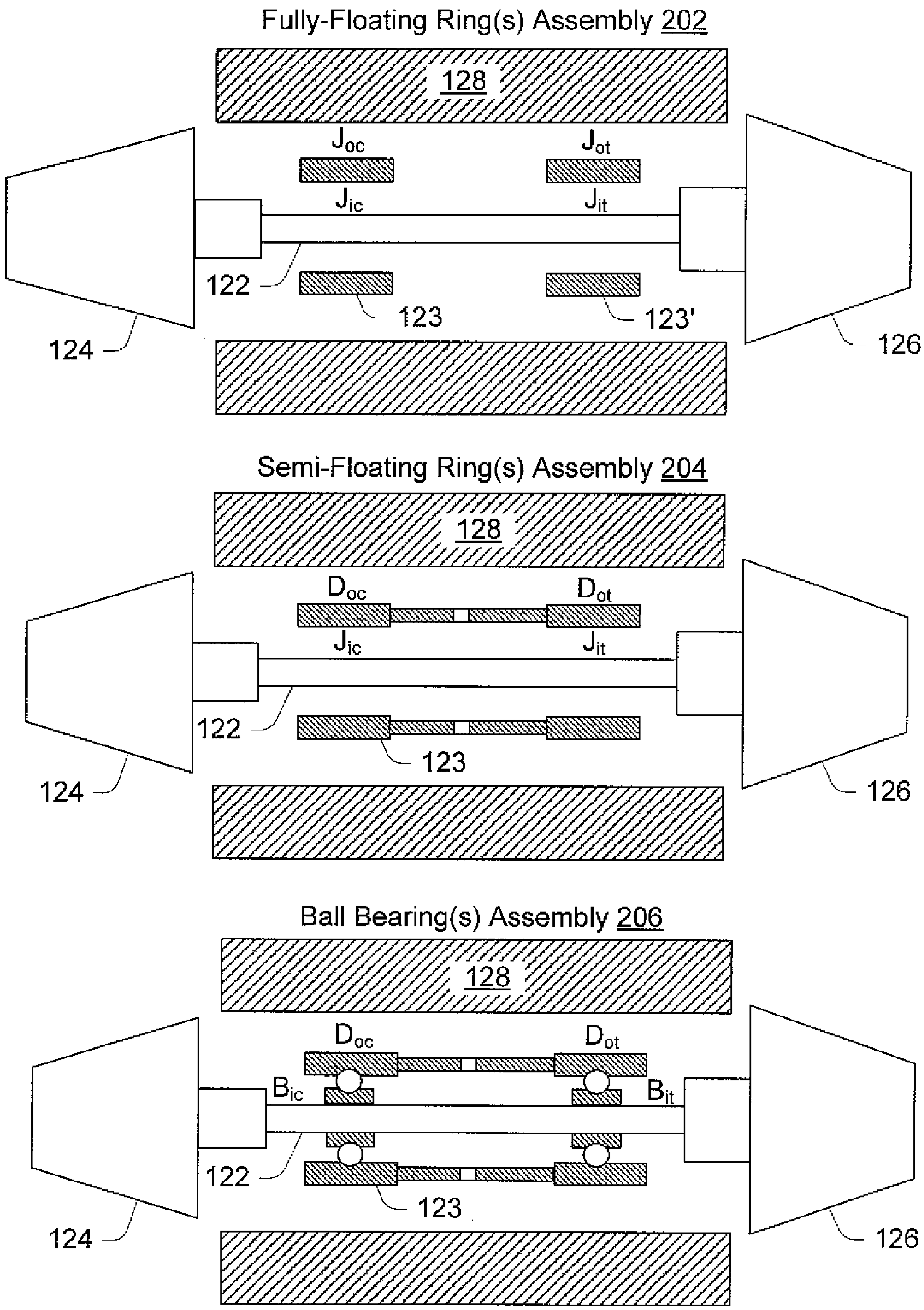


Fig. 2
(Prior Art)

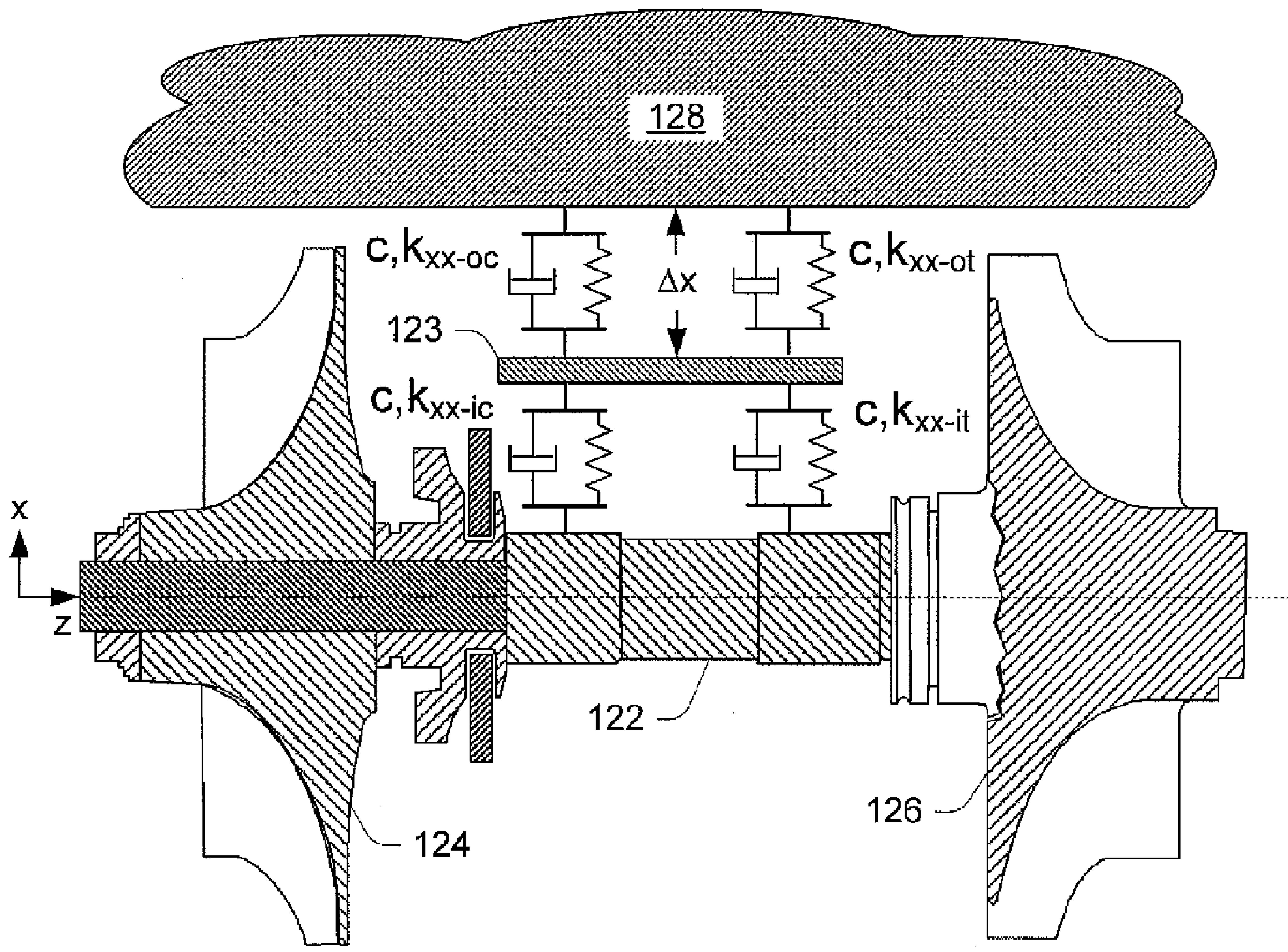


Fig. 3A
(Prior Art)

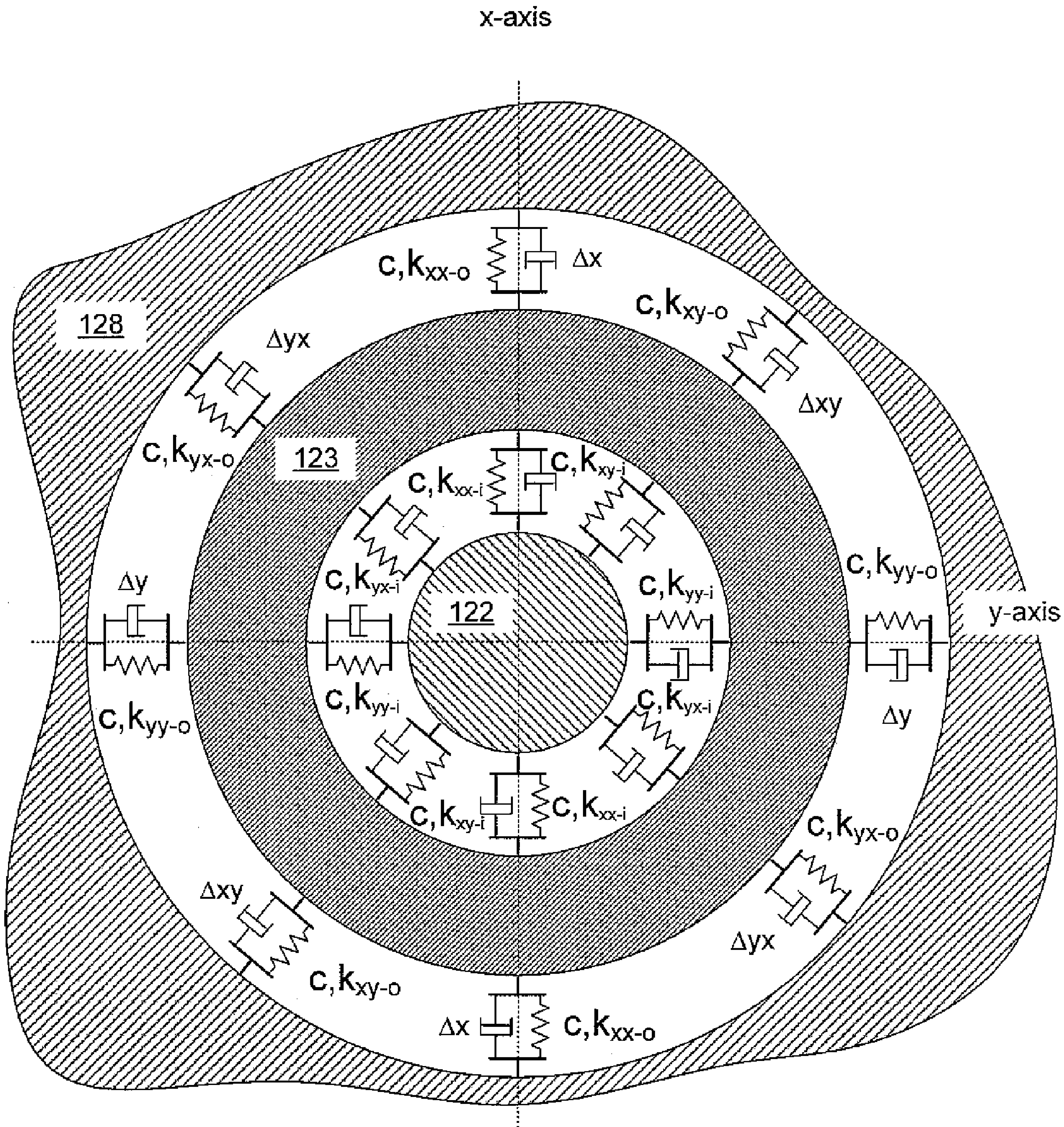


Fig. 3B
(Prior Art)

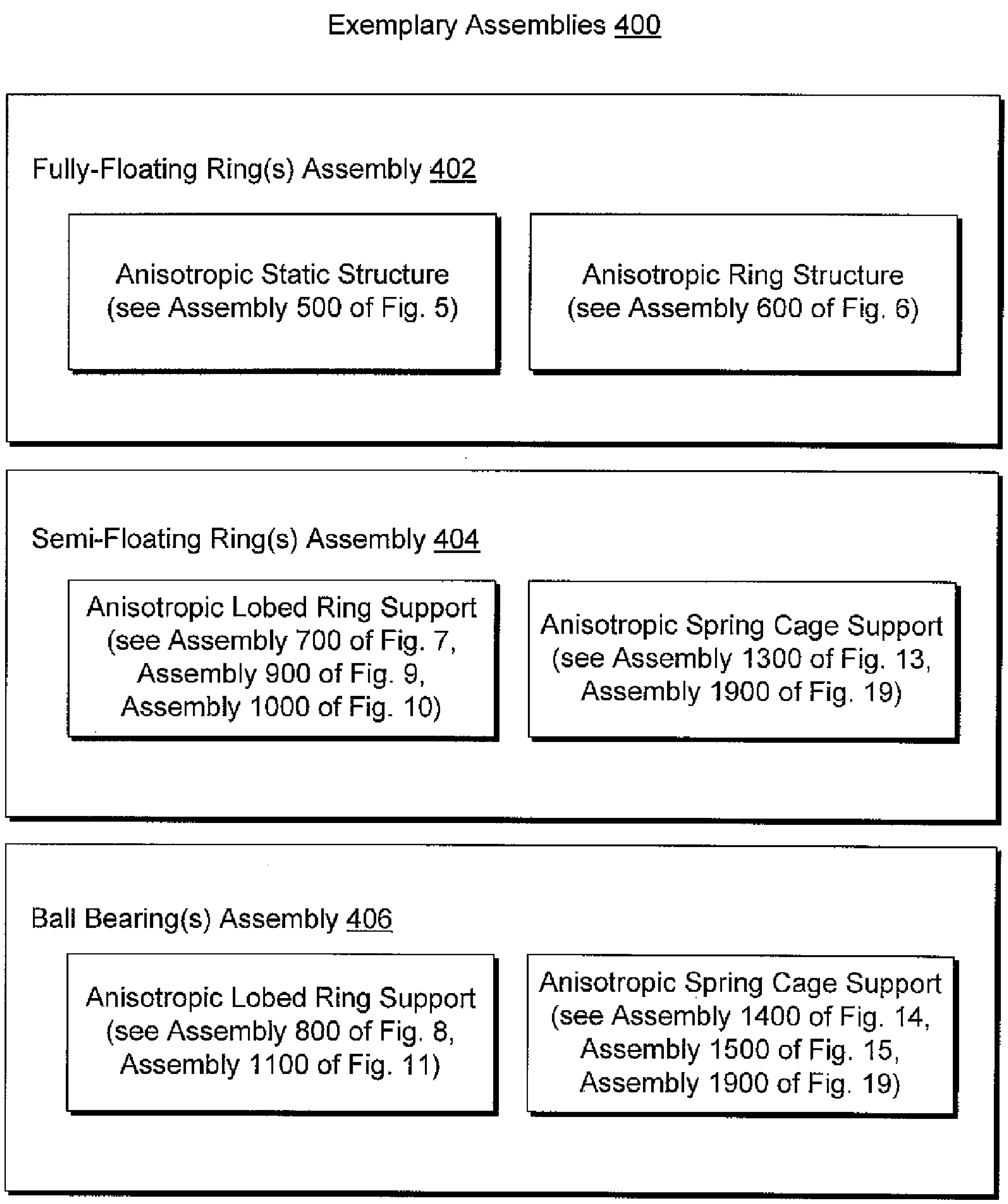

Exemplary Assemblies 400
Fully-Floating Ring(s) Assembly 402
Anisotropic Static Structure (see Assembly 500 of Fig. 5)
Anisotropic Ring Structure (see Assembly 600 of Fig. 6)
Semi-Floating Ring(s) Assembly 404
Anisotropic Lobed Ring Support (see Assembly 700 of Fig. 7, Assembly 900 of Fig. 9, Assembly 1000 of Fig. 10)
Anisotropic Spring Cage Support (see Assembly 1300 of Fig. 13, Assembly 1900 of Fig. 19)
Ball Bearing(s) Assembly 406
Anisotropic Lobed Ring Support (see Assembly 800 of Fig. 8, Assembly 1100 of Fig. 11)
Anisotropic Spring Cage Support (see Assembly 1400 of Fig. 14, Assembly 1500 of Fig. 15, Assembly 1900 of Fig. 19)

Fig. 4

Exemplary Fully-Floating Assembly 500

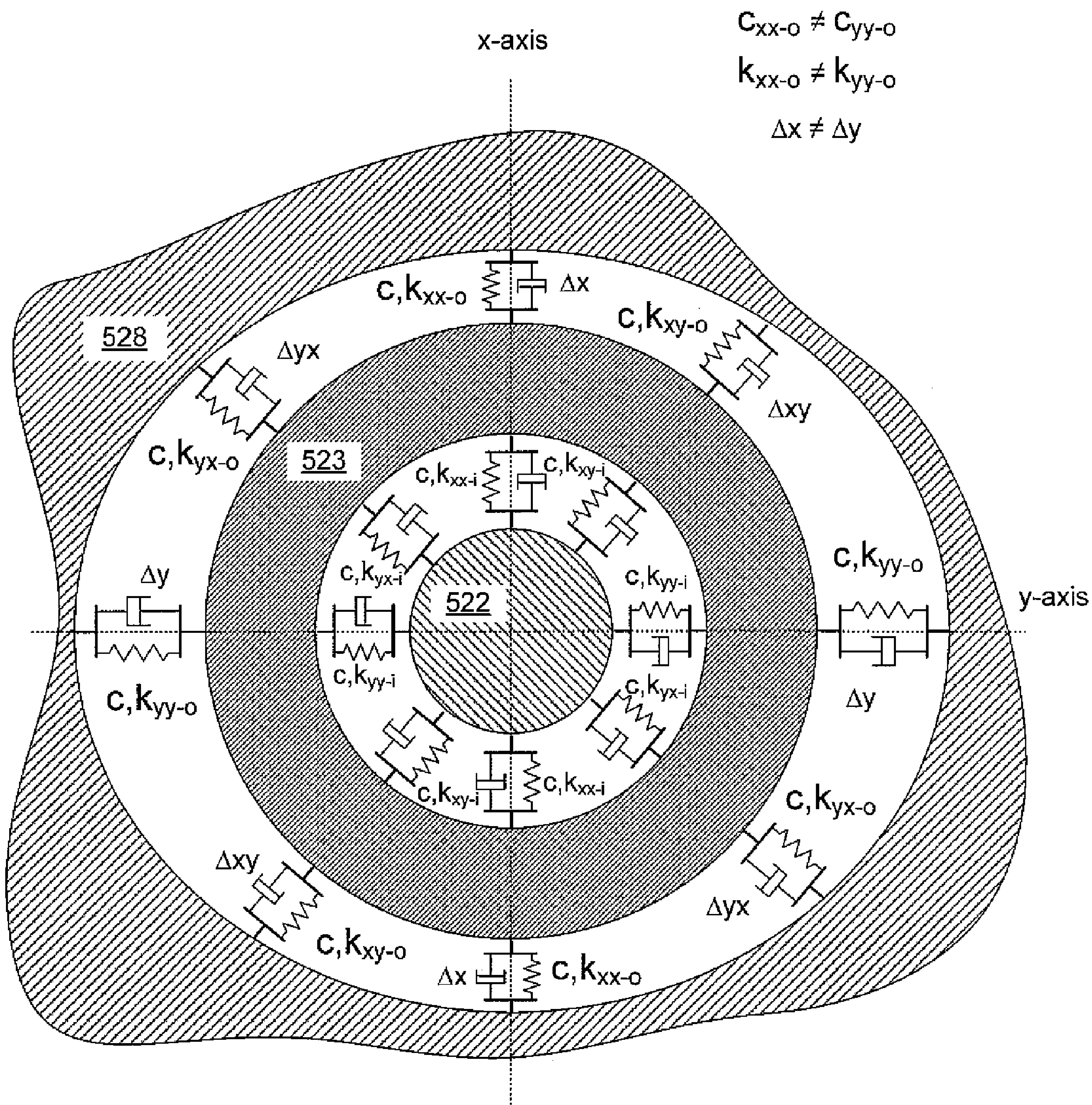


Fig. 5

Exemplary Lobed Ring and Fully-Floating Assembly 600

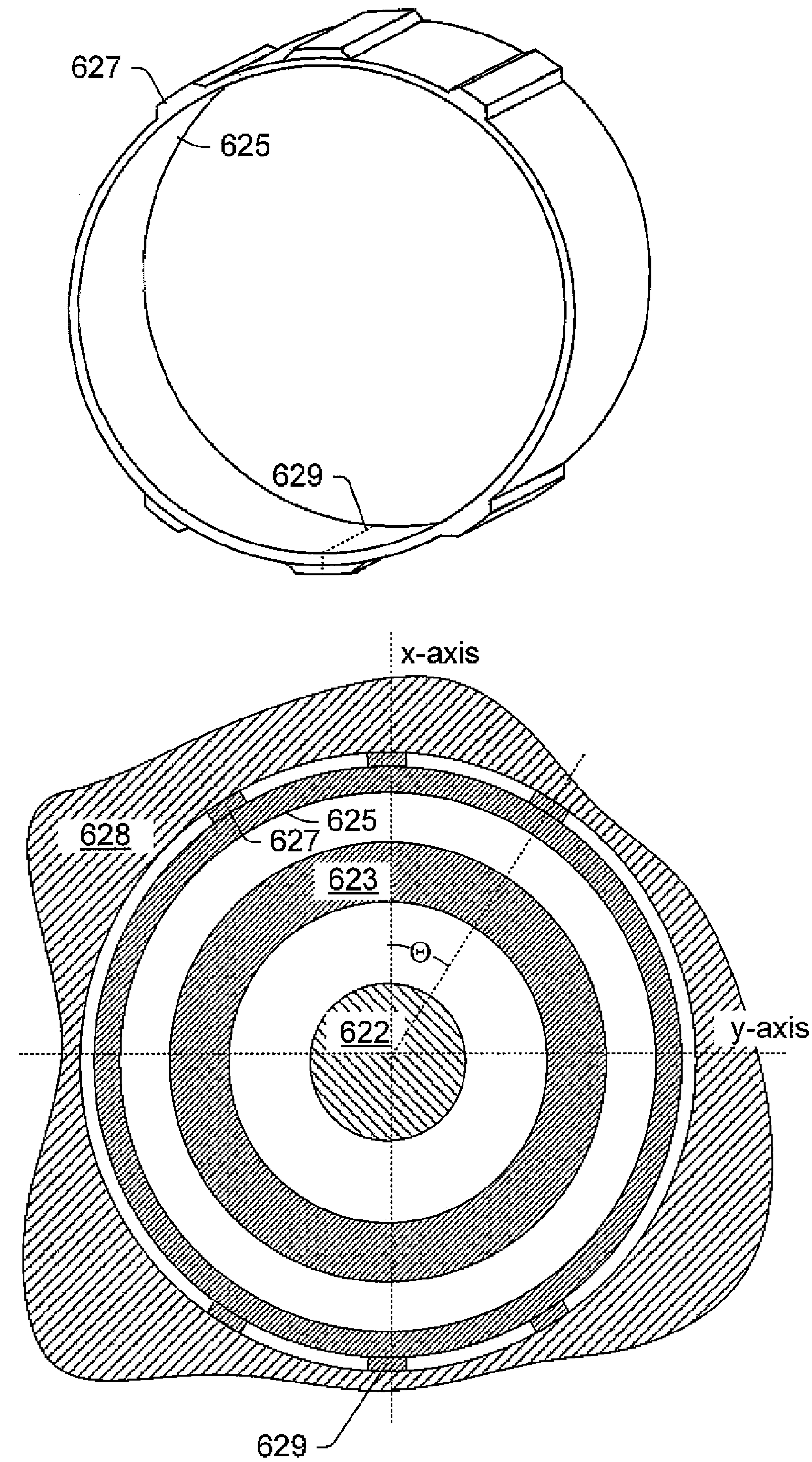


Fig. 6

Exemplary Semi-Floating Assembly with Rings (open) 700

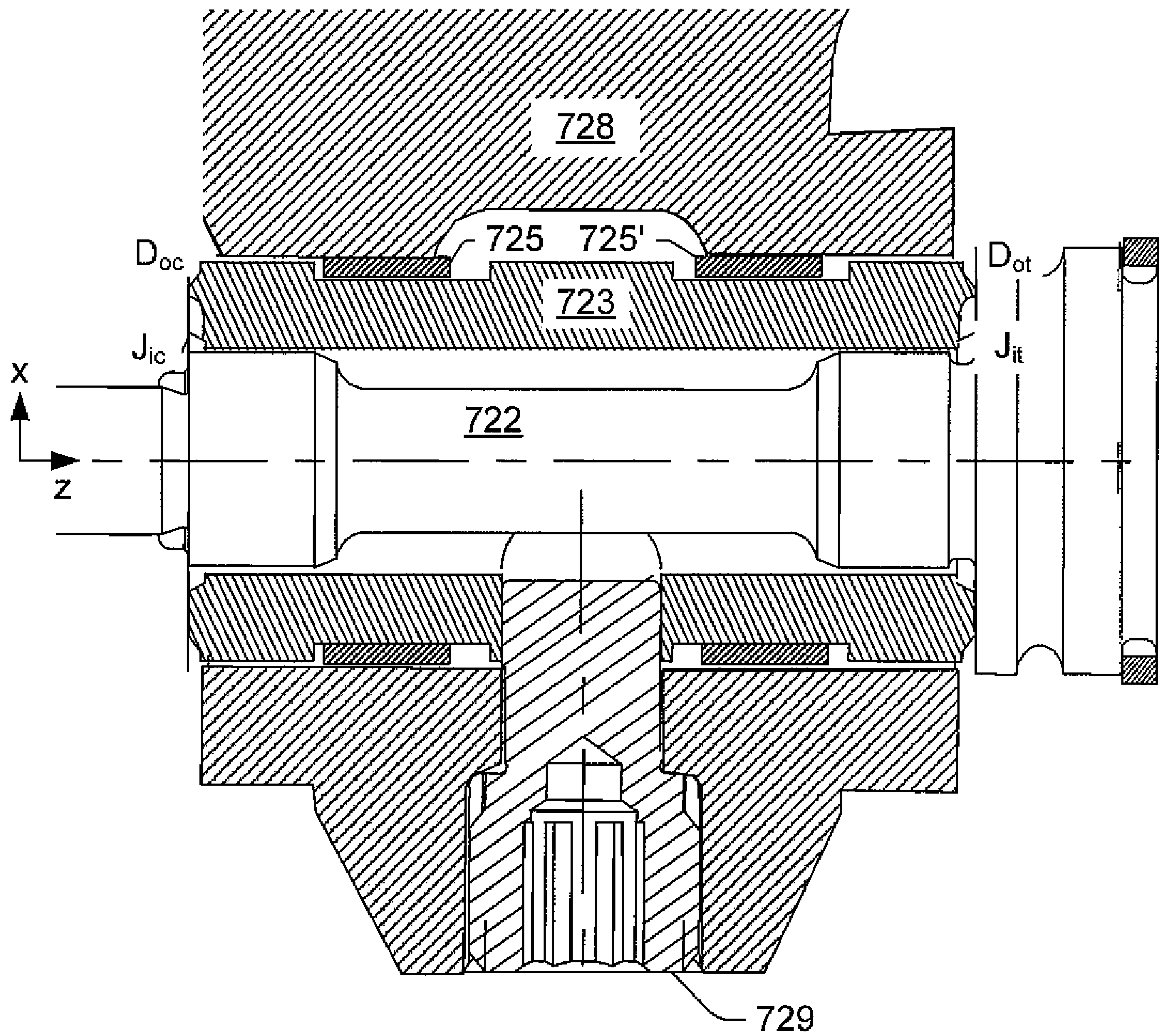


Fig. 7

Exemplary Ball Bearing Assembly with Rings (open) 800

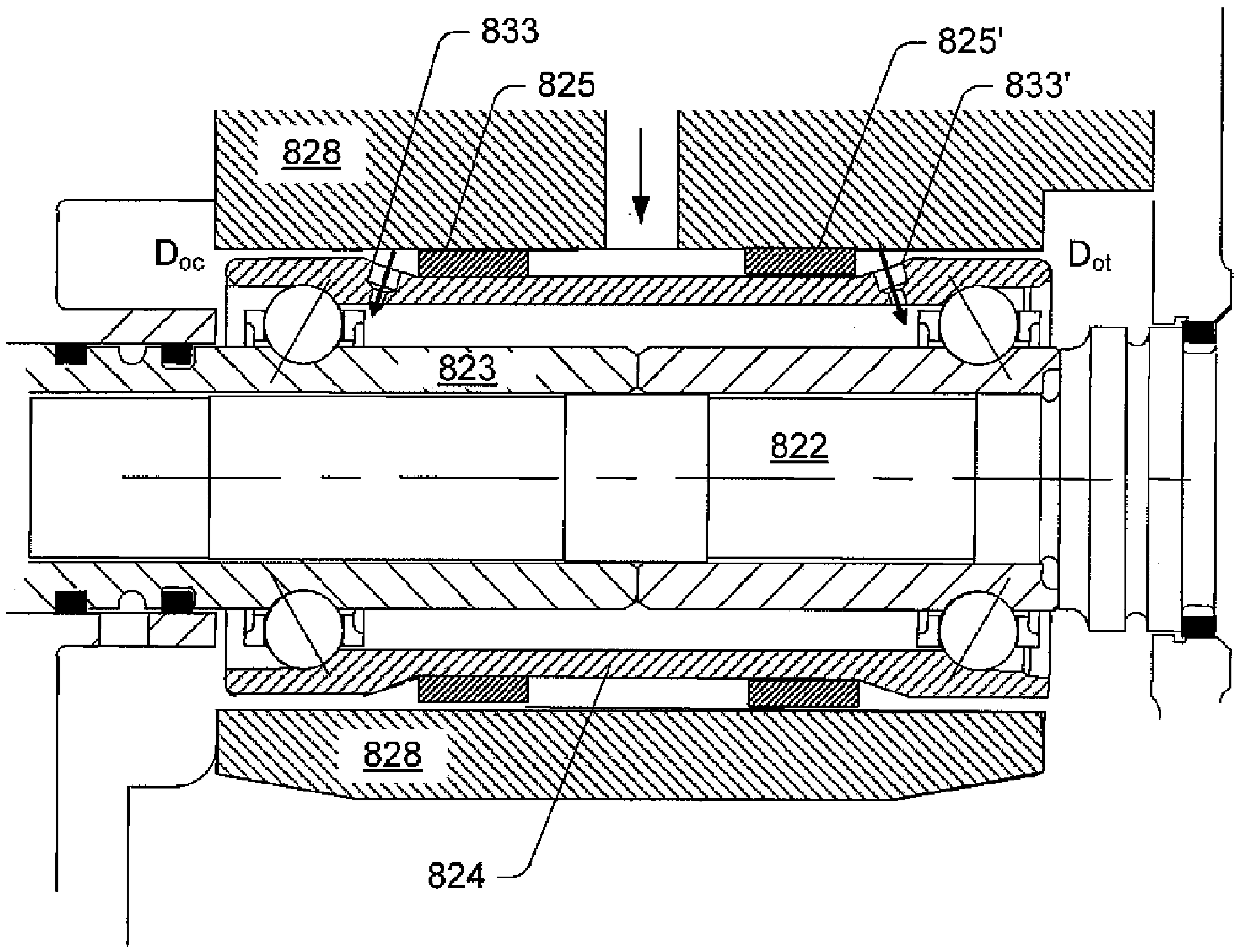


Fig. 8

Exemplary Semi-Floating Assembly with Rings 900

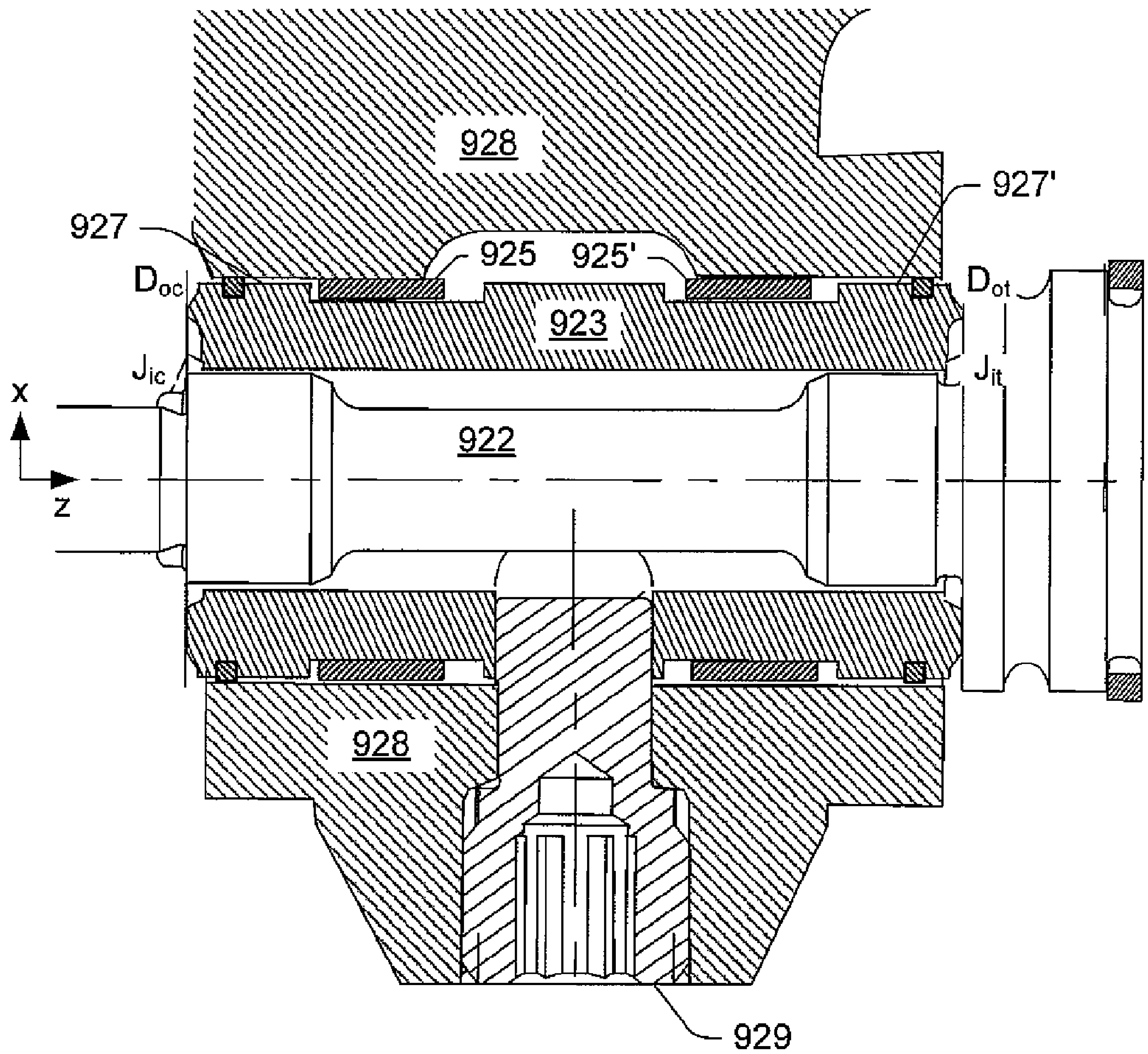


Fig. 9

Exemplary Semi-Floating Assembly with Rings 1000

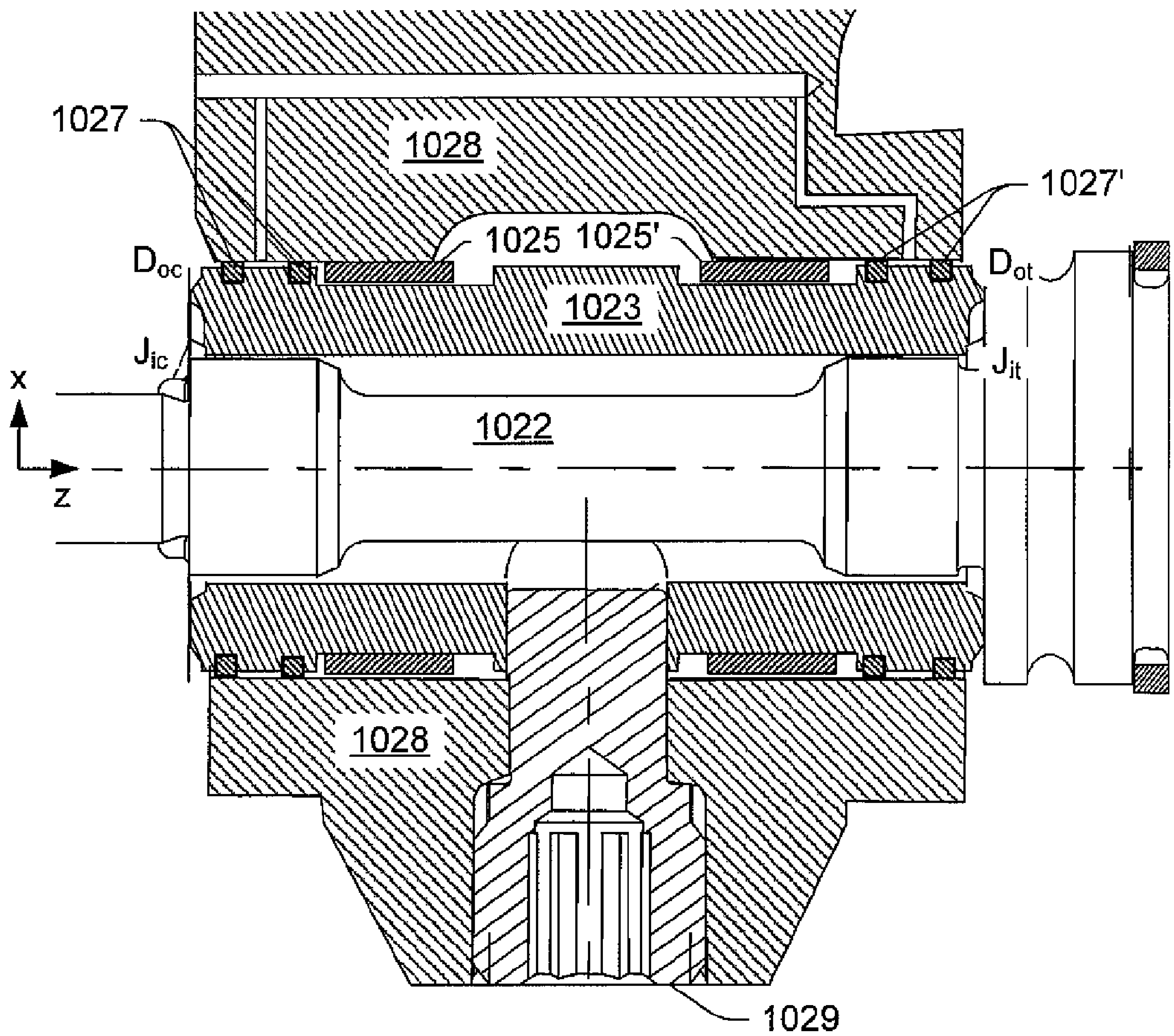


Fig. 10

Exemplary Ball Bearing Assembly with Ring 1100

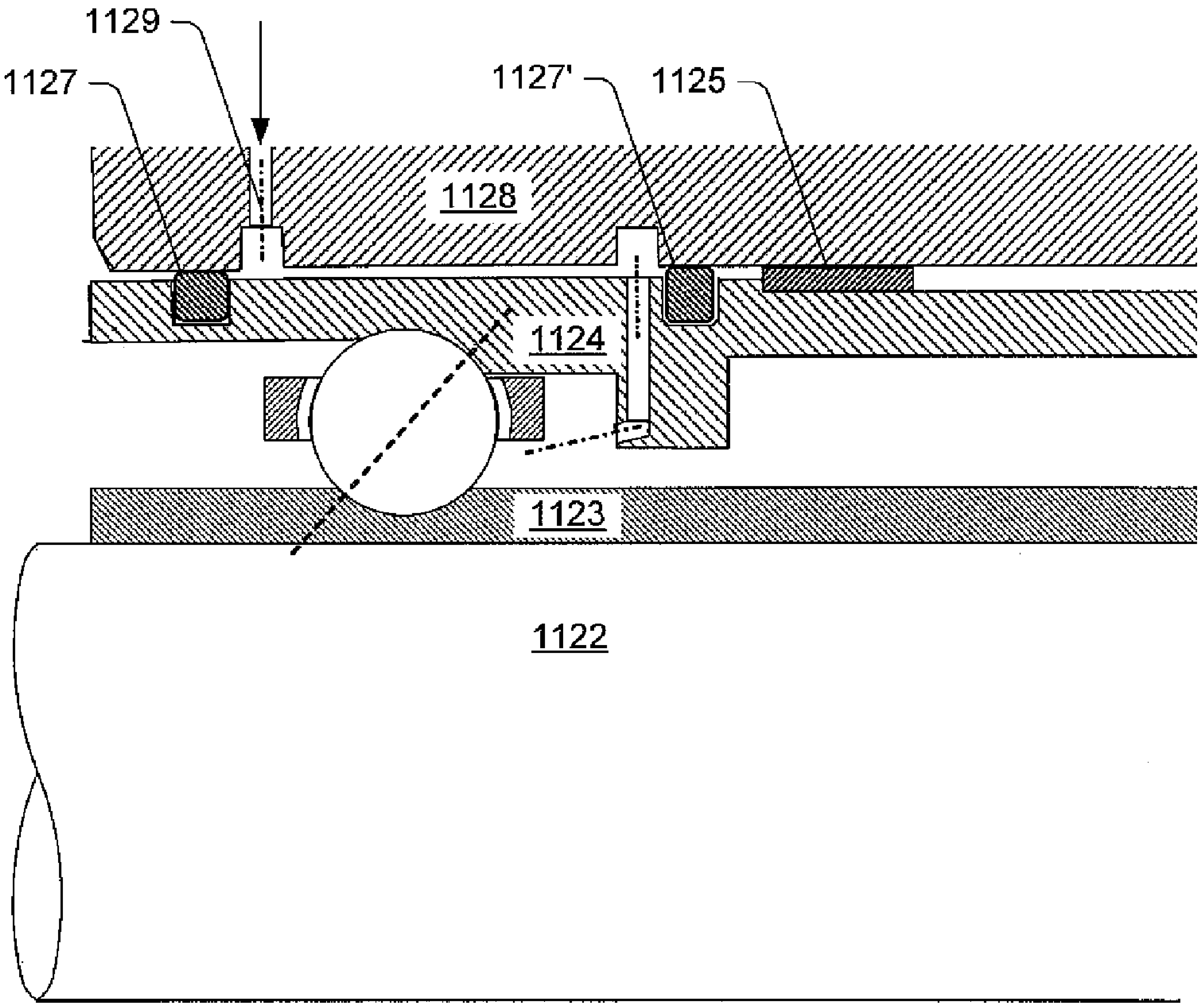


Fig. 11

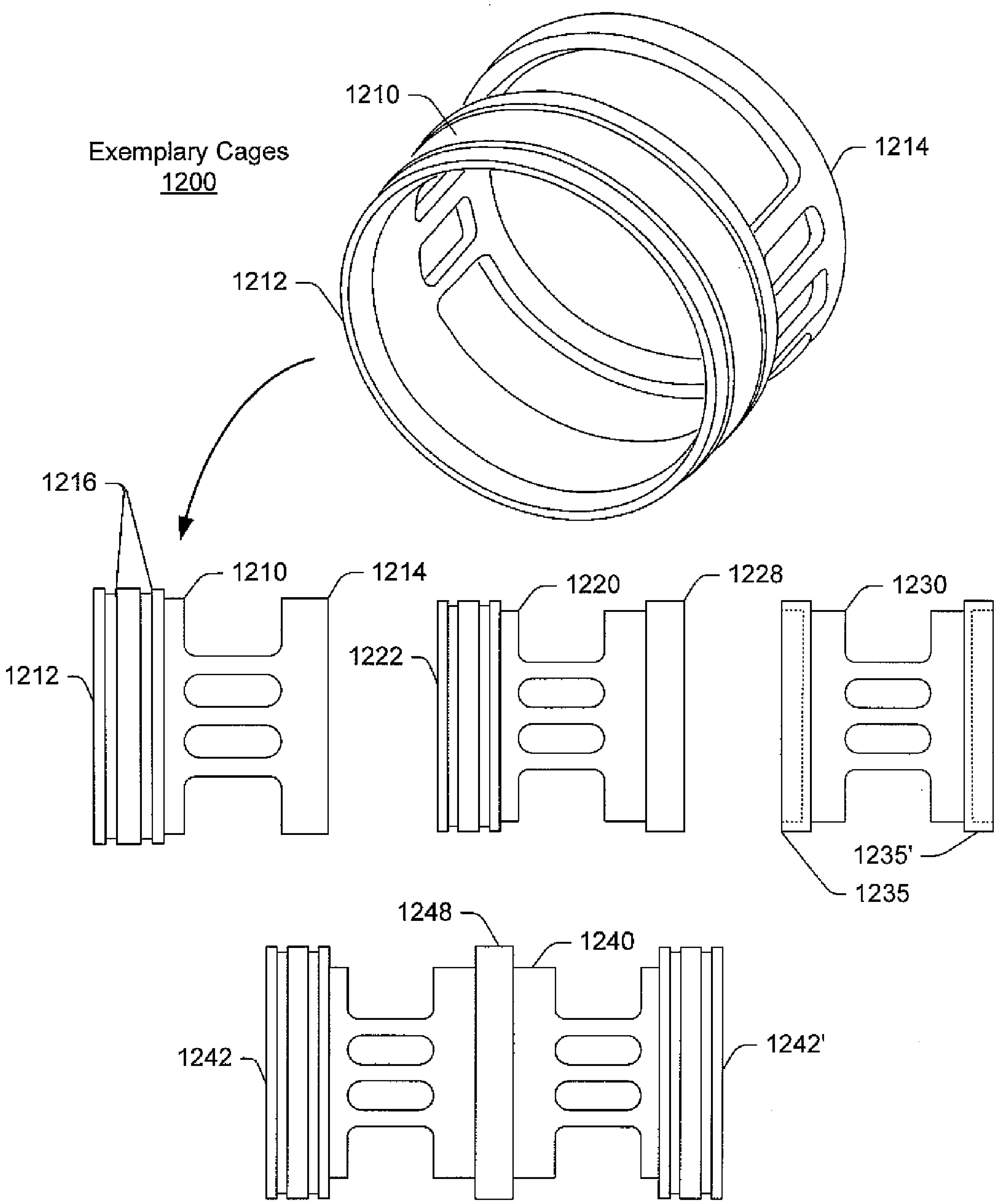

Exemplary Cages
1200
1210
1214
1212
1216
1210
1214
1212
1220
1228
1222
1230
1235'
1235
1248
1240
1242
1242'

Fig. 12

Exemplary Semi-Floating Assembly with Cage 1300

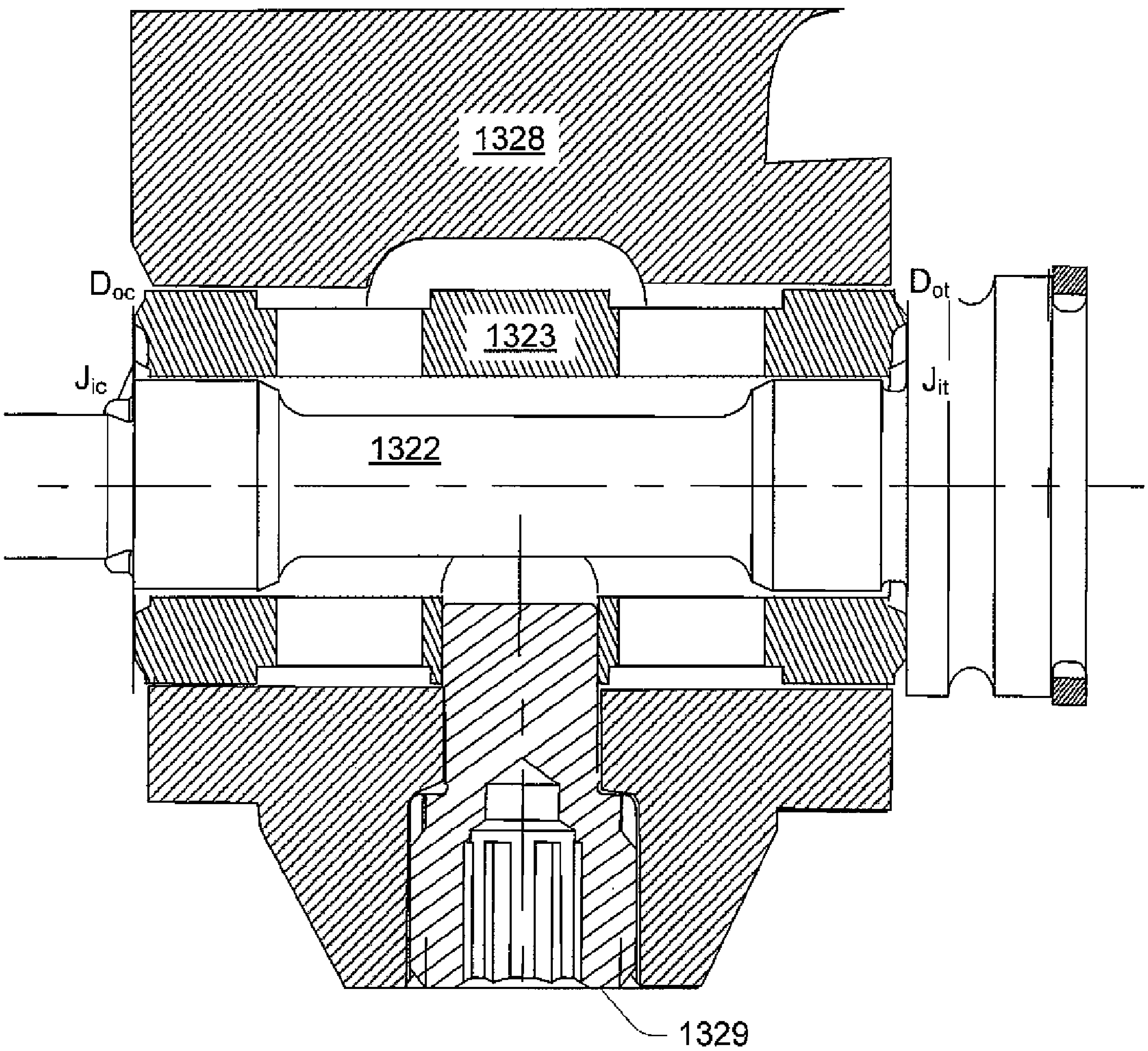


Fig. 13

Exemplary Ball Bearing Assembly with Cage 1400

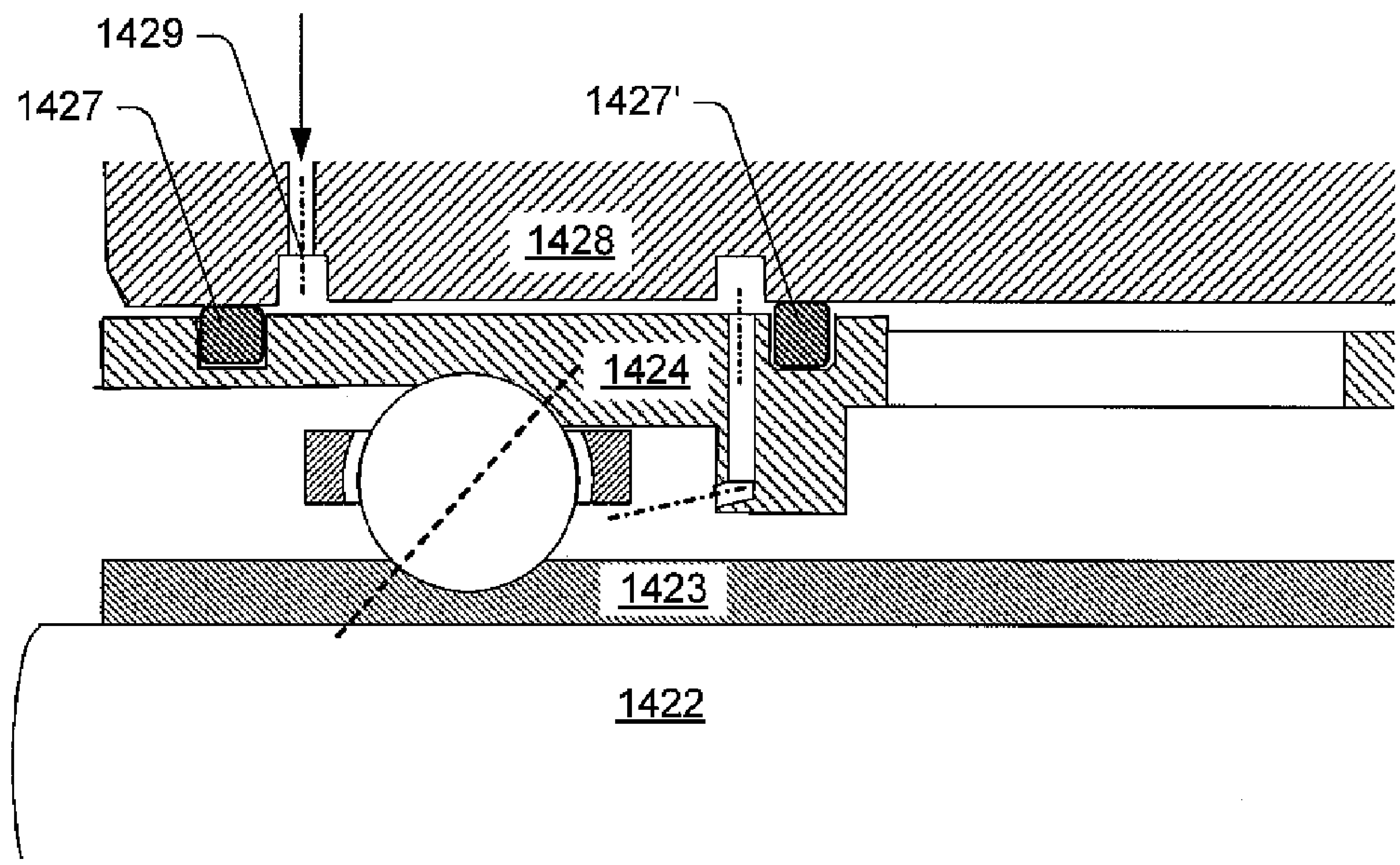


Fig. 14

Exemplary Ball Bearing Assembly with Cage 1500

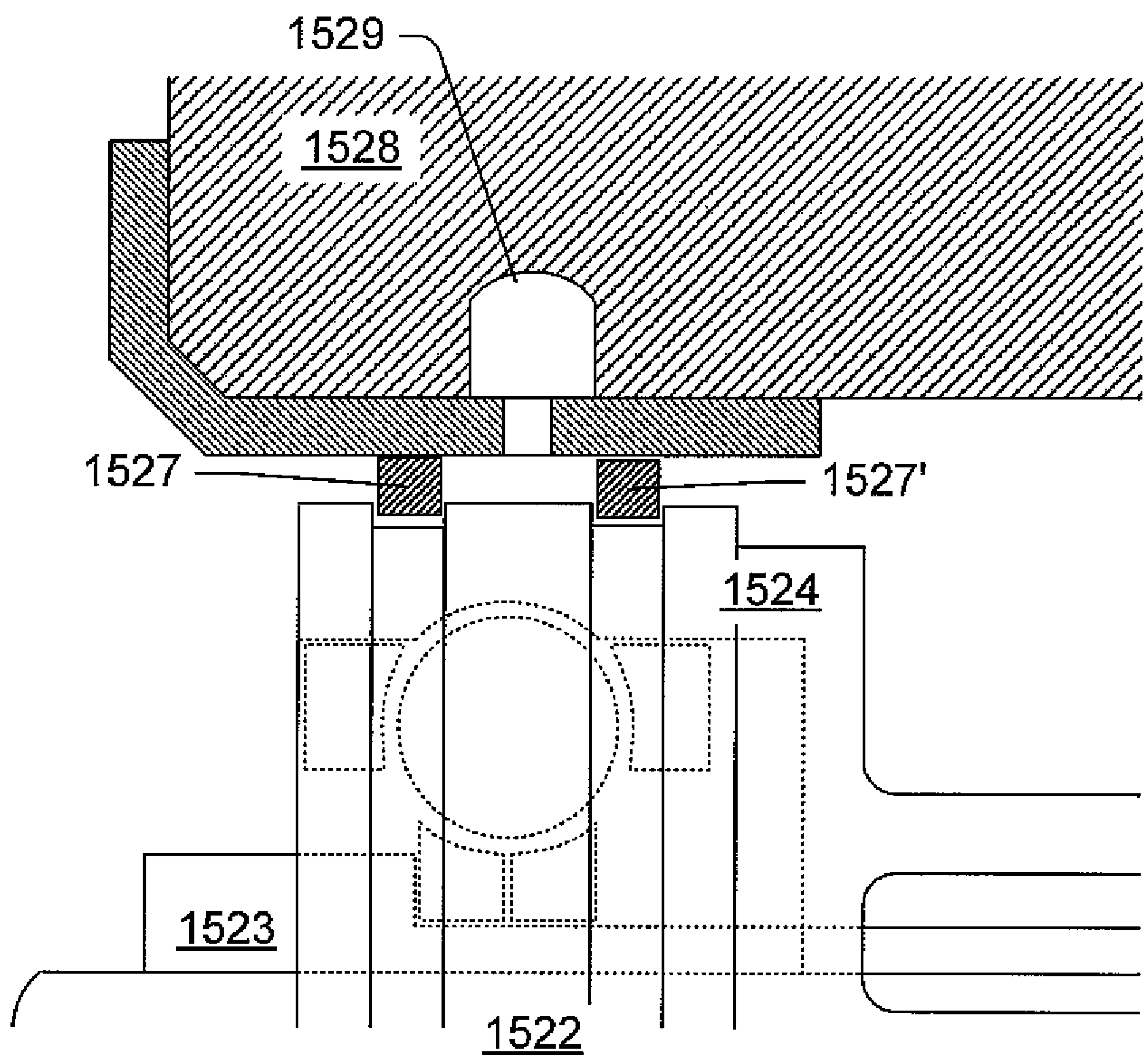


Fig. 15

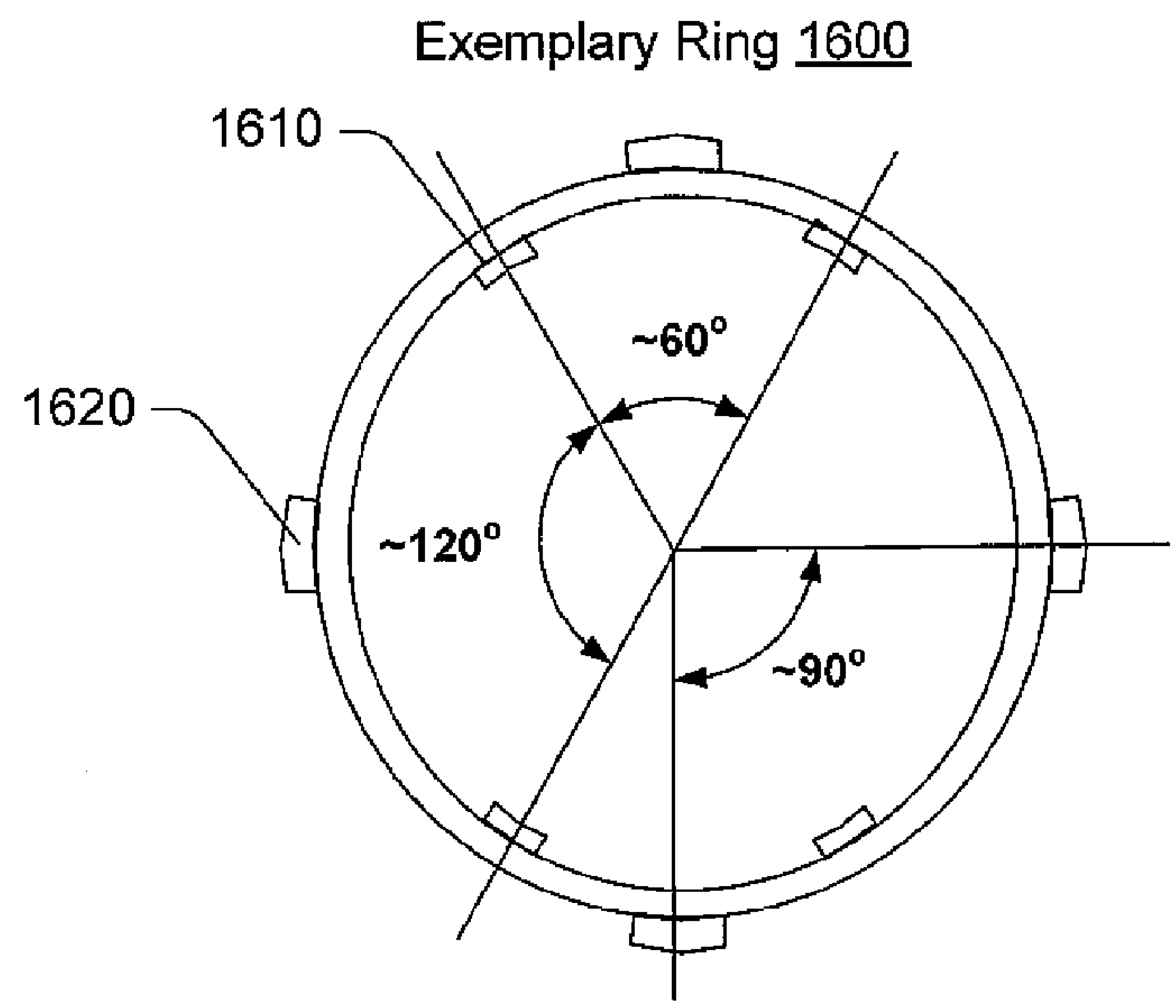

Exemplary Ring 1600
1610
1620
~60°
~120°
~90°

Fig. 16

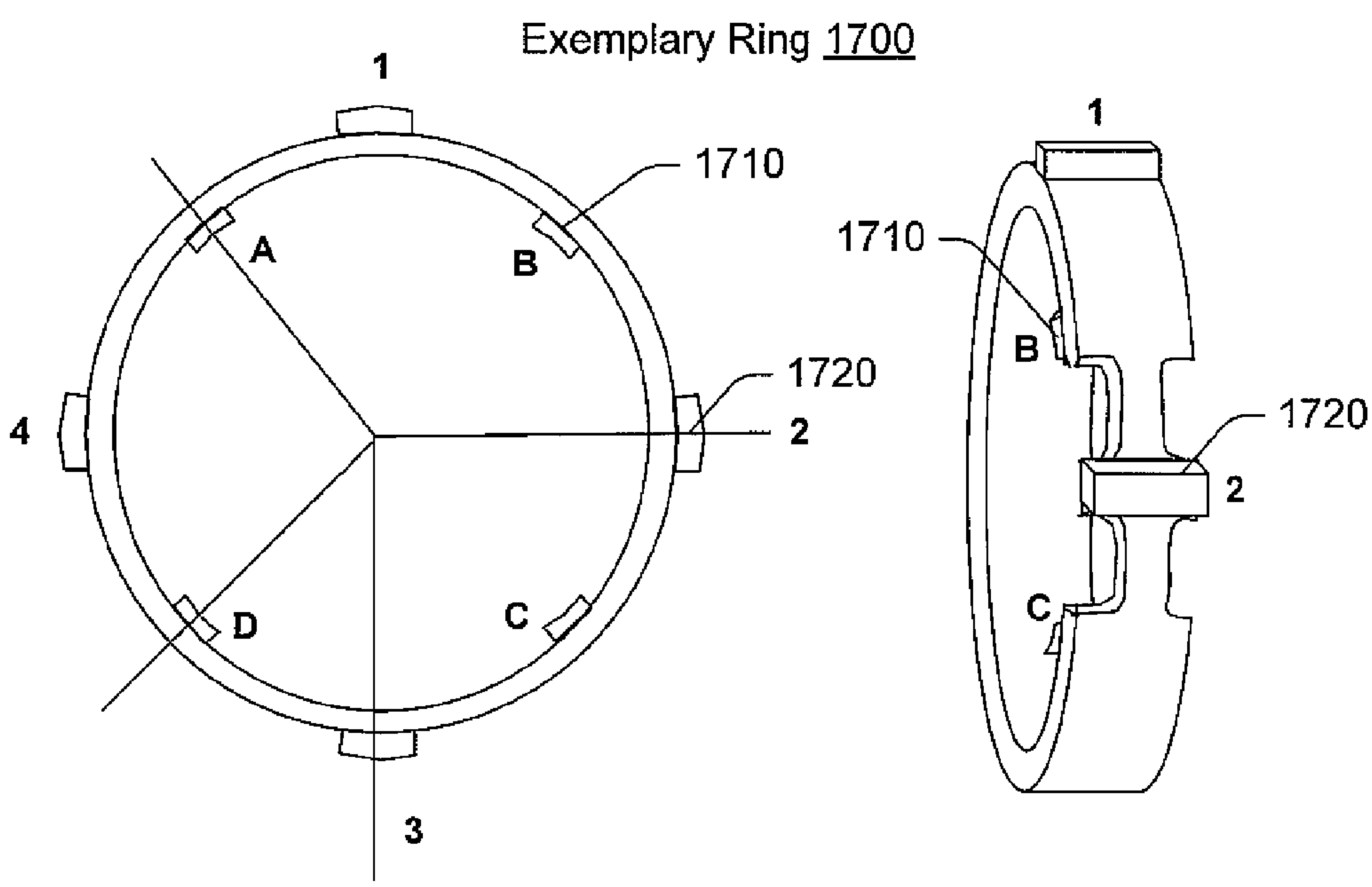

Exemplary Ring 1700
1
1710
A
B
1720
4
2
D
C
3
1
1710
B
1720
2
C

Fig. 17

Exemplary Ring 1800

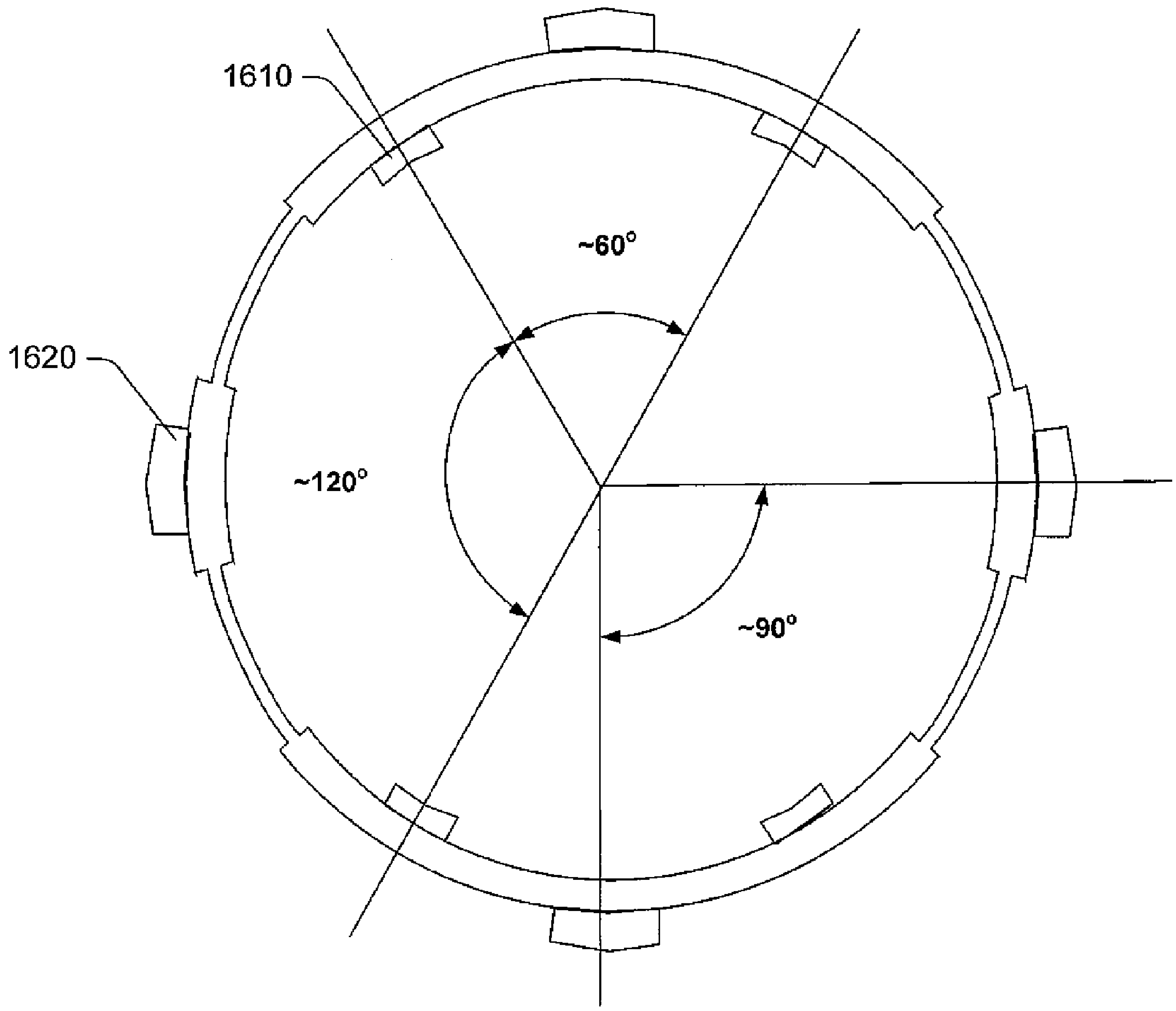


Fig. 18

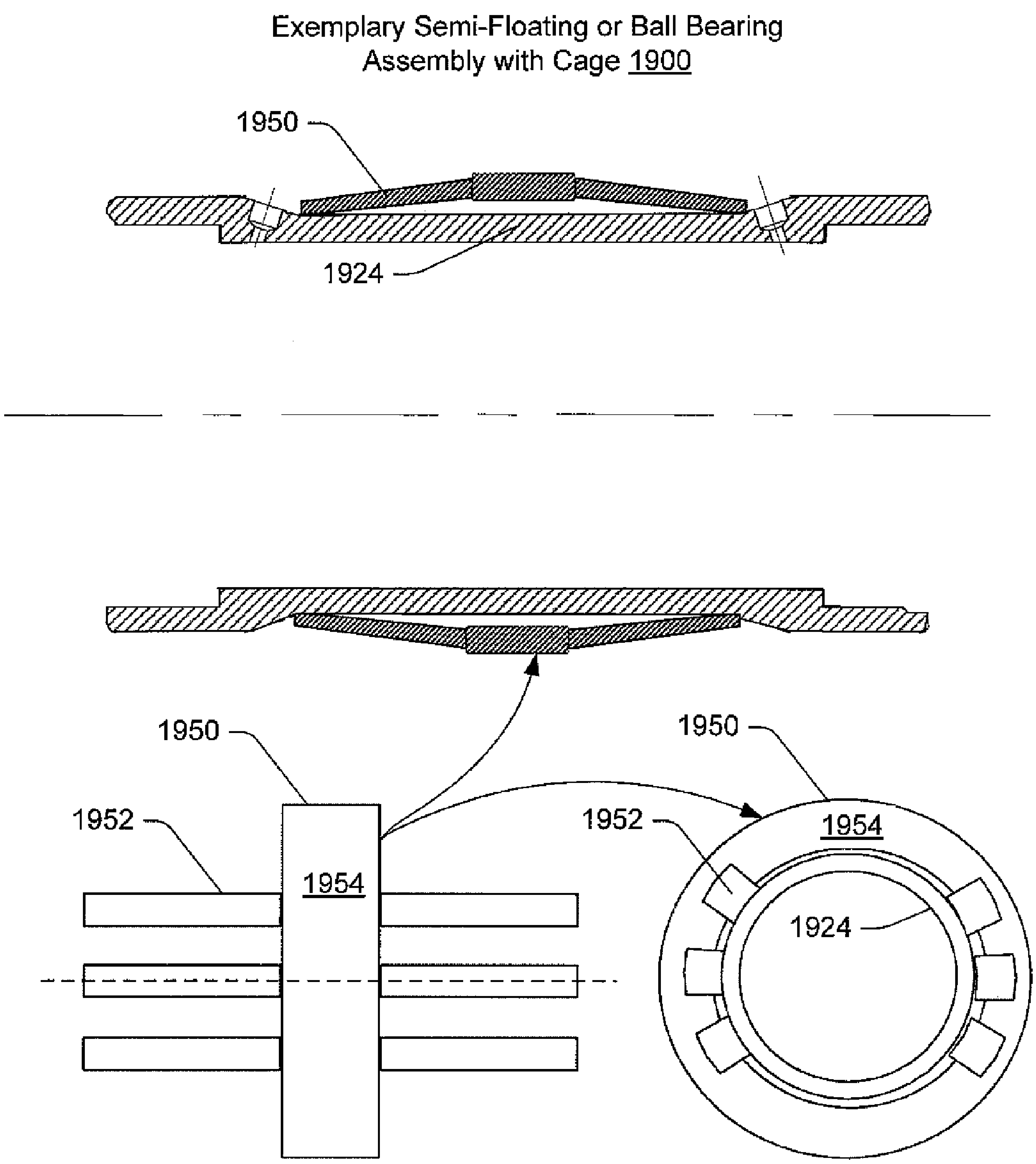

Exemplary Semi-Floating or Ball Bearing
Assembly with Cage 1900
1950
1924
1950
1952
1954
1950
1952
1954
1924

Fig. 19

ANISOTROPIC BEARING SUPPORTS FOR TURBOCHARGERS

RELATED APPLICATIONS

This application is a divisional of U.S. patent application having Ser. No. 11/930,841, filed Oct. 31, 2007 (issued as U.S. Pat. No. 8,118,570), which is incorporated by reference herein.

TECHNICAL FIELD

Subject matter disclosed herein relates generally to turbomachinery for internal combustion engines and, in particular, bearings and components for use with bearings.

BACKGROUND

Turbo-machinery, in general, can experience non-synchronous vibration (NSV). NSV is usually associated with an unstable rotor-bearing system mode, however, in many cases, a limit cycle is reached, which limits the amplitude of NSV. Large amplitude or unbound NSV can result in excessive vibration/noise and/or destruction of the turbo-machinery. NSV which is low frequency (typically lower than the synchronous speed of the machine) can result in undesirable noise.

NSV can be the result of many design parameters. Control of these parameters is not always easy, and in some cases unavoidable due to basic design requirements of a particular turbo-machine. In many instances, NSV control is accomplished by modifying rotor supports or optimization of the supports, often at increased cost, complexity or reduced component tolerance. However, these efforts often do not fully suppress NSV.

Full suppression of NSV for turbochargers that must operate over a large range of speeds, temperatures and external loading is seldom achieved. Depending on design, a turbocharger rotor may be mounted using floating ring bearings or partial floating ring bearings, which have clearances that allow for rotor drop (e.g., due to gravity). A designer typically needs to balance: (i) bearing clearance for rotor stability (minimization of NSV), (ii) rotor clearances for performance and (iii) turbocharger operability. To balance these factors, the operating envelope of the bearings (clearances, oil temperature range, oil type) requires extensive testing to verify that NSV is controlled. However, testing cannot always account for minor changes in bearing clearance due to wear, which can lead to NSV on turbochargers.

Another drawback of conventional turbocharger bearing systems is the large amount of lubricant required for a semi-floating ring bearing supported by a squeeze film damper (SFD) or a ball bearing supported by a SFD (noting that for a fully-floating ring, a lubricant layer lubricates rotation of the ring with respect to a surrounding support structure). SFD systems typically have open mounts that increase lubricant supply requirements to achieve optimum performance. In an alternative "closed" mount approach, sealing and re-use of lubricant results in a reduction of the lubricant required by a turbocharger; which in turn allows for use of a smaller lubricant pump for the engine. Such an approach also leads to an overall reduction in parasitic losses—leading to higher performance vehicles which are more fuel efficient.

Overall, a need exists for bearing technologies that address issues like noise, wear and performance. Various exemplary bearing components and housings presented herein can address such issues.

SUMMARY

An exemplary anisotropic member supports a bearing in a bore and can reduce non-synchronous vibration (NSV) of the bearing in the bore. An exemplary anisotropic member includes an annular body configured to receive a bearing and to space the bearing a distance from a bore surface and is configured to impart anisotropic stiffness and damping terms to the bearing when positioned in the bore. Such a member is suitable for use in a rotating assembly for a turbocharger where the bearing may be a floating bearing, a semi-floating bearing or a ball bearing. Various exemplary members, bearings, housings, assemblies, etc., are disclosed.

BRIEF DESCRIPTION OF THE DRAWINGS

A more complete understanding of the various methods, devices, systems, arrangements, etc., described herein, and equivalents thereof, may be had by reference to the following detailed description when taken in conjunction with the accompanying drawings wherein:

FIG. **1** is a diagram of a conventional internal combustion engine and turbocharger.

FIG. **2** is a series of diagrams for a conventional fully-floating rings assembly, a conventional semi-floating ring assembly, and a conventional ball bearing assembly.

FIG. **3**A is a cross-sectional view of a conventional assembly that includes a bearing positioned in a center housing of a turbocharger.

FIG. **3**B is a cross-sectional view of the conventional assembly of FIG. **3**A.

FIG. **4** is a block diagram listing various exemplary assemblies that include one or more anisotropic features to reduce NSV.

FIG. **5** is a cross-sectional view of an exemplary fully-floating assembly that includes an anisotropic static structure (e.g., an anisotropic center housing bore).

FIG. **6** is a perspective view of an anisotropic structure and a cross-sectional view of an exemplary fully-floating assembly that includes the anisotropic structure positioned within a bore (e.g., a center housing bore).

FIG. **7** is a cross-sectional view of an exemplary semi-floating assembly that includes a pair of anisotropic structures positioned within a bore (e.g., a center housing bore) to center a bearing in the bore.

FIG. **8** is a cross-sectional view of an exemplary ball bearing assembly that includes a pair of anisotropic structures positioned within a bore (e.g., a center housing bore).

FIG. **9** is a cross-sectional view of an exemplary semi-floating assembly that includes a pair of anisotropic structures positioned within a bore (e.g., a center housing bore) to center a bearing in the bore and that includes a pair of piston rings to seal a compressor side SFD and a turbine side SFD.

FIG. **10** is a cross-sectional view of an exemplary semi-floating assembly that includes a pair of anisotropic structures positioned within a bore (e.g., a center housing bore) to center a bearing in the bore and that includes two pairs of piston rings where one pair seals a compressor side SFD and another pair seals a turbine side SFD.

FIG. **11** is a cross-sectional view of an exemplary ball bearing assembly with an anisotropic structure and a pair of piston rings to seal a SFD.

FIG. **12** is a perspective view of an exemplary anisotropic cage structure along with a side view of the cage and various side views of alternative cages.

FIG. **13** is a cross-sectional view of an exemplary semi-floating assembly that includes an anisotropic cage.

FIG. **14** is a cross-sectional view of an exemplary ball bearing assembly with an anisotropic cage along with a pair of piston rings that define a lubricant film region.

FIG. **15** is a cross-sectional view of an exemplary ball bearing assembly with an anisotropic cage that seats a pair of piston rings that define a lubricant film region.

FIG. **16** is a plan view of an exemplary anisotropic structure that includes a plurality of segments.

FIG. **17** is a plan view and a perspective view of an exemplary anisotropic structure that includes a plurality of segments where one or more segments differ in width (e.g., Δz).

FIG. **18** is a plan view of an exemplary anisotropic structure that includes a plurality of segments where one or more segments differ in thickness (e.g., Δr).

FIG. **19** is a cross-sectional view of an exemplary semi-floating or ball bearing assembly with an anisotropic cage.

DETAILED DESCRIPTION

Various exemplary methods, devices, systems, arrangements, etc., disclosed herein address issues related to technology associated with turbochargers and are optionally suitable for use with electrically assisted turbochargers, turbine generators and/or motorized compressors.

Turbochargers are frequently utilized to increase the output of an internal combustion engine. Referring to FIG. **1**, a prior art system **100**, including an internal combustion engine **110** and a turbocharger **120** is shown. The internal combustion engine **110** includes an engine block **118** housing one or more combustion chambers that operatively drive a shaft **112**. As shown in FIG. **1**, an intake port **114** provides a flow path for air to the engine block **118** while an exhaust port **116** provides a flow path for exhaust from the engine block **118**.

The turbocharger **120** acts to extract energy from the exhaust and to provide energy to intake air, which is combined with fuel to form combustion gas. As shown in FIG. **1**, the turbocharger **120** includes an air inlet **134**, a shaft **122**, a compressor **124**, a turbine **126**, a housing **128** and an exhaust outlet **136**. The housing **128** may be referred to as a center housing as it is disposed between the compressor **124** and the turbine **126**. The shaft **122** may be a shaft assembly that includes a variety of components.

Referring to the turbine **126**, such a turbine optionally includes a variable geometry unit and a variable geometry controller. The variable geometry unit and variable geometry controller optionally include features such as those associated with commercially available variable geometry turbochargers (VGTs). Commercially available VGTs include, for example, the GARRETT® VNT™ and AVNT™ turbochargers, which use multiple adjustable vanes to control the flow of exhaust across a turbine. A turbocharger may employ wastegate technology as an alternative or in addition to variable geometry technology.

FIG. **2** shows three different, conventional assembly arrangements for turbocharger rotor support: a fully-floating rings assembly **202**, a semi-floating ring assembly **204** and a ball bearing assembly **206**. Each assembly includes a compressor **124** and a turbine **126** connected by a shaft **122** supported within a static structure **128** (e.g., a center housing). An assembly may include additional rotating components for sealing the compartment ("closed" lubricant arrangement) or for reacting axial load.

The fully-floating rings assembly **202** includes a shaft **122** supported by a set of fully-floating journal bearings **123** (compressor side) and **123'** (turbine side). In the assembly **202**, each side has an outer journal (J_o) facing the static structure **128** and an inner journal (J_i) facing the shaft **122**. Each fully-floating ring **123**, **123'** is supported by an outer lubricant layer and an inner lubricant layer and can rotate about the rotor axis (center-line) as well as move radially (e.g., off-center from the rotor axis). However, each ring **123**, **123'** is typically constrained from moving axially. In the arrangement **202**, each bearing has a cross-coupling term with the other bearing; these terms, by definition, are not anisotropic.

In the semi-floating ring bearing assembly **204**, a ring **123** is constrained from rotating about the rotor axis. For example, one or more pins may prevent or limit rotation of the ring **123**. By limiting or constraining rotation, the outer hydraulic mount for the semi-floating ring **123** becomes a SFD (or SFDs) and the ring **123** is only able to whirl. The assembly **204** includes an inner compressor side journal (J_{ic}), an inner turbine side journal (J_{it}), an outer compressor side damper (D_{oc}) and an outer turbine side damper (D_{ot}).

For the assembly **204**, the inner-journals have cross-coupling terms and are not anisotropic while the outer SFD(s) (e.g., D_{oc} and D_{ot}) have no cross-coupling, however, the normal direction stiffness and damping are equal for center circular response and as such, are not anisotropic by design.

The assembly **206** includes "ball", or more generally, rolling element bearings (B_{ic} and B_{it}) supported by outer SFDs (e.g., D_{oc} and D_{ot}). As shown in FIG. **2**, the assembly **206** includes a rotor **122** supported by a cartridge **123** that includes a set of roller bearings. The cartridge **123** is supported by at least a compressor end SFD (D_{oc}) and a turbine end SFD (D_{ot}). The outer race of the bearing cartridge **123** is similar the semi-floating ring; it can whirl about the rotor centerline but it cannot rotate about the bearing centerline. This results in the outer-hydraulic mount being SFD. For the assembly **206**, the rolling element bearings are isotropic as is the compressor side SFD and the turbine side SFD (no cross-coupling terms and the normal terms are equal). Such rotor-bearing systems (RBS) are generally designed without a centering mechanism and without anisotropic support characteristics.

FIGS. **3**A and **3**B show two different cross-sectional views of an assembly where a compressor side bearing and a turbine side bearing are represented by a series of normal stiffness (k) and damping (c) terms as well as cross-coupling terms. FIG. **3**A shows a rotor assembly that includes a compressor **124** and a turbine **126** coupled to a shaft **122**. A ring **123** (or rings) supports the shaft **122** in the static structure **128** (e.g., a center housing).

A coordinate system is defined with the centerline (rotational axis) of the assembly along a z-axis and an x-axis perpendicular to the z-axis and aligned, for example, with gravity; a y-axis is defined as out of the page and perpendicular to the z-axis and the x-axis. FIG. **3**B shows a cross-sectional view of the assembly in a xy-plane normal to the z-axis. Various terms "c,k" are defined with respect to the coordinate system and proximity to the compressor side or the turbine side of the assembly (e.g., $c,k_{xx\text{-}oc}$, $c,k_{xx\text{-}ic}$, $c,k_{xx\text{-}ot}$, $c,k_{xx\text{-}it}$, etc.). In both FIGS. **3**A and **3**B, the clearance between the ring **123** and the static structure **128** is designed to be approximately equal along the x-axis and along the y-axis (i.e., $\Delta x = \Delta y$). As shown in FIG. **3**B, neglecting gravity, the outer terms (between the ring **123** and the static structure **128**) are substantially equal and the inner terms (between the ring **123** and the shaft **122**) are substantially equal.

As mentioned in the Background section, in a conventional assembly where the terms are substantially equal, NSV can occur. As described herein, an exemplary assembly includes anisotropy to suppress NSV.

FIG. **4** shows a chart of exemplary assemblies **400** that correspond to assemblies **500**, **600**, **700**, **800**, **900**, **1000**,

1100, **1300**, **1400**, **1500** and **1900** of FIGS. **5**, **6**, **7**, **8**, **9**, **10**, **11**, **13**, **14**, **15** and **19** respectively. The assemblies **400** are grouped as fully-floating **402**, semi-floating **404** and ball bearing **406** (or other similar rotating element).

The fully-floating rings assembly **402** may be anisotropic by introduction of anisotropy in a static structure (see assembly **500** of FIG. **5**) and/or by introduction of anisotropy in a ring structure (see assembly **600** of FIG. **6**). The semi-floating ring assembly **404** may be an assembly made anisotropic by introduction of anisotropy in a ring structure (see assembly **700** of FIG. **7**, assembly **900** of FIG. **9** and assembly **1000** of FIG. **10**) or it may be an assembly made anisotropic by introduction of an anisotropic cage (see assembly **1300** of FIG. **13** and assembly **1900** of FIG. **19**). The ball bearing assembly **406** may be an assembly made anisotropic by introduction of anisotropy in a ring structure (see assembly **800** of FIG. **8** and assembly **1100** of FIG. **11**) or it may be an assembly made anisotropic by introduction of an anisotropic cage (see assembly **1400** of FIG. **14**, assembly **1500** of FIG. **15** and assembly **1900** of FIG. **19**).

While the assemblies of FIG. **3** show both a compressor and a turbine, in other examples, an anisotropic structure may be used with a half assembly, i.e., an assembly with a compressor only or a turbine only. In addition, an anisotropic structure may be used with a motorized turbocharger, a motorized compressor and/or a turbine coupled to a generator. In addition, one of the assemblies of FIG. **3** might include only a compressor bearing supported by an anisotropic member or only a turbine bearing supported by an anisotropic member. In such examples, the other bearing (or bearings) case could be unsupported or supported by an isotropic support.

FIG. **5** shows a cross-sectional view of an assembly **500** with an anisotropic static structure **528**. The cross-sectional view may be at a position along the z-axis at the compressor side, at the turbine side or at a point between the compressor side and the turbine side of the assembly **500**. The static structure **528** can be a center housing with an anisotropic bore. For example, a center housing bore with an elliptical cross-section is anisotropic with respect to a ring with a circular cross-section.

The assembly **500** includes a conventional ring **523** and a conventional shaft **522** positioned in the static structure **528**. The ring **523** can rotate about the centerline (i.e., z-axis) as supported by an outer lubricant layer between the ring **523** and the static structure **528** and an inner lubricant layer between the ring **523** and the shaft **522**. As indicated, the outer layer term k_{xx-o} is not equal to the outer layer term k_{yy-o}, the outer layer term c_{xx-o} is not equal to the outer layer term c_{yy-o} and the clearance Δx between the ring **523** and the static structure **528** is not equal to the clearance Δy between the ring **523** and the static structure **528**; hence, the assembly **500** is anisotropic with respect to the outer layer that supports the ring **523** and the shaft **522**. This type of anisotropy acts to suppress NSV and enhance rotor stability. More specifically, the ring **523** rotates in an anisotropic lubricant space that suppresses NSV that would occur in an isotropic lubricant space.

According to the fully-floating rotating ring example of FIG. **5**, an anisotropic support is one in which the stiffness and damping terms are designed such that c, k_{yy} are not equal to c, k_{xx} and where the cross coupling terms are approximately zero (c,k_{yx}, c,k_{xy}=0).

As described herein, for a substantially non-rotating support as found in a semi-floating assembly and a ball bearing assembly, an outer journal and/or a center housing bore can be machined such that a clearance in one direction (e.g., vertical) differs from a clearance in another direction (e.g., horizontal).

In the examples of FIG. **6** through FIG. **19**, anisotropy is introduced by an anisotropic support member positioned in a static structure bore, which may be a substantially isotropic bore. In general, an anisotropic support member does not rotate to any significant degree and hence rotational dynamics of such an anisotropic support is not an important design consideration. Further, lubricant management features such as piston rings can reduce turbocharger lubricant flow requirements. Various features can be incorporated individually or in conjunction.

FIG. **6** shows an exemplary fully-floating assembly **600** that includes an anisotropic support structure **625** positioned between a ring **623** and a static structure **628**. In this example, the anisotropic structure **625** affects interactions between the outer journal surface of the ring **623** and the static structure **628**. The anisotropic structure **625** includes a series of spacers **627** that are positioned at various angles (θ) about the centerline (z-axis) to generate some amount of asymmetry. Where the anisotropic support **625** is resilient, the spacers **627** alter the stiffness terms for the outer journal surface of the ring **623**. The anisotropic structure **625** optionally includes a split **629** to facilitate assembly. For example, the ring structure **625** can be opened at the split **629** and positioned around a bearing (e.g., the ring **623**).

The anisotropic support structure **625** can be a lobed spring that biases the static structure **628**. While the anisotropic support structure **625** has a generally isotropic shape, except for the spacers **627**, the structure **625** can have an anisotropic shape that acts to reduce NSV.

While a center housing bore for a turbocharger may have some anisotropy, such anisotropy is typically for lubricant drainage, ease of assembly, etc., and not to control NSV. As described herein, particular components of a center housing for a turbocharger can include anisotropic features that reduce NSV. For example, various examples pertain to bearing supports that include anisotropic characteristics to control NSV.

FIG. **7** shows a semi-floating assembly **700** that includes two anisotropic support members **725**, **725'** positioned between a semi-floating bearing **723** and a static structure **728**. In this example, the static structure **728** is a center housing of a turbocharger that includes a bore and an aperture configured to accept a locating pin **729** that extends into the bore to axially locate and limit rotation of the bearing **723** about the centerline of the bore (e.g., z-axis). Clearance between the pin **729** and the bearing **723** allow the bearing **723** to move along the x-axis.

The bearing **723** includes an outer, compressor side film damper surface (D_{oc}) and an outer, turbine side film damper surface (D_{ot}) that define, with the bore of the static structure **723**, an outer lubricant film thickness for a compressor side SFD and a turbine side SFD, respectively. Similarly, the shaft **722** includes an inner, compressor side journal surface (J_{ic}) and an inner, turbine side journal surface (J_{it}) that define, with the bore of bearing **723**, an inner, compressor side lubricant film thickness and an inner, turbine side lubricant film thickness, respectively. Noting that the shaft **722** rotates at high speed with respect to the constrained semi-floating bearing **723**; thus, the inner lubricant films experience much more shear than the outer SFDs. As the outer SFDs are not bound on the compressor side or the turbine side, lubricant flow to maintain the SFDs may be substantial. In other words, such an open configuration tends to be sensitive to lubricant flow and lubricant pressure.

The anisotropic support members **725**, **725'** can be springs (or act as springs) to bias and center the bearing **723** in the bore, which may reduce or eliminate bearing drop along the x-axis in instances where lubricant pressure falls (e.g., engine shut down). Such an arrangement can reduce bearing instabilities associated with fluctuations or gradients in lubricant pressure.

In general, the members **725**, **725'** are lobed or bump springs are used to center the ring **723**. An anisotropic bump spring can include inner and/or outer lobes with uneven spacing, similar to the spacers **627** of the structure **625** of FIG. **6**.

FIG. **8** shows a ball bearing assembly **800** that includes a pair of anisotropic centering springs **825**, **825** for centering a bearing cartridge that includes a two piece inner race **823** and a unitary outer race **824**. The compressor side SFD and the turbine side SFD are formed by damper surfaces D_{oc} and D_{ot} of the bearing outer race **824** and a bore surface of the static structure **828**. The unitary outer race of the bearing cartridge **824** includes lubricant jet apertures **833**, **833'**, which form part of the lubricant system.

FIG. **9** shows a semi-floating assembly **900** that includes a pair of anisotropic centering springs **925**, **925'** for centering a ring **923**. In the example of FIG. **9**, seal rings **927**, **927'** seal a compressor side SFD and a turbine side SFD, respectively. The seal rings **927**, **927'** can help to reduce lubricant requirements for the assembly **900**. Further, each of the springs **925**, **925'** can help define a respective SFD boundary. Various other features are described with respect to the example of FIG. **8**.

FIG. **10** shows an exemplary semi-floating assembly **1000** that includes a pair of anisotropic centering springs **1025**, **1025'** for centering a ring **1023**. In the example of FIG. **10**, a compressor side pair of seal rings **1027** seal a compressor side SFD and a turbine side pair of seal rings **1027'** seal a turbine side SFD. The pairs of seal rings **1027**, **1027'** can help to reduce lubricant requirements for the assembly **1000**. In the example of FIG. **10**, lubricant paths exist in the static structure **1029** to deliver lubricant to the compressor side SFD and to the turbine side SFD. Various other features are described with respect to the example of FIG. **8**.

FIG. **11** shows an exemplary ball bearing assembly **1100** that includes an anisotropic ring **1125**. The ball bearing assembly **1100** includes a ball bearing that includes an inner race **1123** mounted on a shaft **1122** and an outer race **1124** that forms a SFD with a static structure **1128**. The static structure **1128** includes a lubricant passage **1129** that delivers lubricant to the SFD where the SFD is defined in part by an outboard seal ring **1127** and an inboard seal ring **1127'**.

FIG. **12** shows various exemplary anisotropic cages **1200**. An exemplary cage **1210** includes an outboard end **1212** and an inboard end **1214**. In this example, the outboard end **1212** includes one or more grooves or channels **1216** where each groove or channel can receive a seal ring.

In another example, an anisotropic cage **1220** includes an outboard end **1222** and an inboard end **1228** that can form an SFD with a static structure. In yet another example, an anisotropic cage **1230** includes symmetric ends **1235**, **1235'** that can receive, for example, respective bearings. In another example, an anisotropic cage **1240** includes two cage portions separated by a support portion **1248** where the cage has symmetric ends **1242**, **1242'**.

FIG. **13** shows an exemplary semi-floating assembly **1300** with an anisotropic cage **1323** located in a static structure **1328** to support a shaft **1322**. The anisotropic cage **1323** includes an inner compressor side journal surface J_{ic}, an inner turbine side journal surface J_{it}, an outer compressor side film damper surface D_{oc} and an outer turbine side film damper surface D_{ot}. A pin **1329** locates the cage **1323** in the static structure **1328**. The cage **1323** may have one or more features of one or more of the cages **1200** of FIG. **12**. In particular, the cage **1323** includes an opening to receive the pin **1329** to limit rotation of the cage **1323** in the static structure **1328**.

FIG. **14** shows an exemplary ball bearing assembly **1400** that includes an anisotropic cage **1424** that acts as an outer race. Hence, the ball bearing assembly **1400** includes a ball bearing that includes an inner race **1423** mounted on a shaft **1422** and an anisotropic cage outer race **1424** that forms a SFD with a static structure **1428**. The static structure **1428** includes a lubricant passage **1429** that delivers lubricant to the SFD where the SFD is defined in part by an outboard seal ring **1427** and an inboard seal ring **1427'**. The cage **1423** may have one or more features of one or more of the cages **1200** of FIG. **12**.

FIG. **15** shows an exemplary ball bearing assembly **1500** that includes an anisotropic cage **1524** that acts as an outer race such as the cage **1230** of FIG. **12** (e.g., one half of the cage **1230**). Hence, the ball bearing assembly **1500** includes a ball bearing that includes an inner race **1523** mounted on a shaft **1522** and an anisotropic cage outer race **1524** that forms a SFD with a static structure **1528**. The static structure **1528** includes a lubricant passage **1529** that delivers lubricant to the SFD where the SFD is defined in part by an outboard seal ring **1527** and an inboard seal ring **1527'**.

FIG. **16** shows an exemplary anisotropic ring **1600** that includes a series of inner spacers **1610** and a series of outer spacers **1620**. The spacers **1610**, **1620** can create anisotropy in the ring **1600** by defining segments of different arc lengths. For example, the inner spacers **1610** define two segments of arc angle ~60° and two segments of arc angle ~120°. The outer spacers **1620** define four segments of arc angle ~90°. Additional anisotropy can be introduced by altering the position and/or number of outer spacers **1620**. In general, different arc length segments will respond differently to force where a shorter arc length is stiffer than a longer arc length.

FIG. **17** shows an exemplary anisotropic ring **1700** that includes a series of inner spacers **1710** and a series of outer spacers **1720**. The spacers **1710**, **1720** define segments of equal arc length. However, the width (Δz) varies for particular segments to create anisotropy. In general, different width segments will respond differently to force where a wider segment is stiffer than a narrower segment.

FIG. **18** shows an exemplary anisotropic ring **1800** that includes a series of inner spacers **1810** and a series of outer spacers **1820**. The spacers **1810**, **1820** can create anisotropy in the ring **1800** by defining segments of different arc lengths. For example, the inner spacers **1810** define two segments of arc angle ~60° and two segments of arc angle ~120°. The outer spacers **1820** define four segments of arc angle ~90°. Additional anisotropy is introduced by differing segment thickness (e.g., Δr). In general, different segments will respond differently to force where a thicker segment is stiffer than a thinner segment. In addition, the ring **1800** can have anisotropy even though the arc angles between all of the spacers **1810** and **1820** are 45° because segments of the ring **1800** have differing segment thicknesses.

FIG. **19** shows an exemplary semi-floating or ball bearing assembly **1900** with an anisotropic cage **1950**. The cage **1950** is positioned with respect to a semi-floating ring or outer race **1924** to support the semi-floating ring or outer race **1924** in a static structure (not shown in FIG. **19**). The cage **1950** includes legs **1952** that extend from a ring portion **1954** where the legs **1952** are arranged in a manner to make the cage anisotropic. In an alternative, the legs **1950** may be of different material, length, thickness, width, etc., to make the cage **1950** anisotropic. Further, the ring portion **1954** optionally includes one or more spacers to make the cage **1950** anisotropic. The ring portion **1950** optionally includes a split to allow for opening of the ring for ease of assembly with respect to a semi-floating ring or outer race.

Various other sealing methods and spring designs can reduce NSV and/or reduce lubricant requirements. The exemplary bearing features discussed herein can reduce design and development time. Such features can diminish bearing wear, which typically leads to NSV. Thus, NSV can be reduced or eliminated during the life cycle of an assembly. Reduction in the amount of lubricant required can increase overall system performance and operability.

As mentioned, various anisotropic members can be used with electrically assisted turbochargers. For example, a rotating assembly for a electrically assisted turbocharger can include a center housing that includes a through bore having a central axis, a pair of bearings positioned in the through bore, an anisotropic member positioned in the through bore between one bearing of the pair of bearings and a surface of the center housing where the surface defines at least part of the through bore of the center housing, a shaft, rotatably supported in the center housing by the pair of bearings, the shaft connected at one end to a compressor wheel and at another end to a turbine wheel and an electric motor configured to drive the shaft. Such an arrangement optionally includes another anisotropic member positioned in the through bore between the other bearing of the pair of bearings and a surface of the center housing where the surface defines at least part of the through bore of the center housing. In electric assist arrangements, an electric motor may function as a generator configured to generate electrical energy from exhaust energy.

Although some exemplary methods, devices, systems arrangements, etc., have been illustrated in the accompanying Drawings and described in the foregoing Detailed Description, it will be understood that the exemplary embodiments disclosed are not limiting, but are capable of numerous rearrangements, modifications and substitutions without departing from the spirit set forth and defined by the following claims.

What is claimed is:

1. A rotating assembly for a turbocharger comprising:

a center housing that comprises a through bore having a central axis;

a semi-floating bearing positioned in the through bore;

a locating pin for limiting rotation of the semi-floating bearing in the through bore;

an anisotropic member positioned in the through bore between the semi-floating bearing and a surface of the center housing wherein the surface defines at least part of the through bore of the center housing; and

a shaft, rotatably supported in the center housing by the semi-floating bearing, the shaft connected at one end to a compressor wheel and at another end to a turbine wheel.

2. The rotating assembly of claim **1** wherein the semi-floating bearing comprises a radial groove and a piston ring seated in the groove that forms a seal between the semi-floating bearing and a surface of the center housing wherein the surface defines at least part of the through bore.

3. The rotating assembly of claim **1** comprising two anisotropic members positioned in the through bore of the center housing.

4. The rotating assembly of claim **1** wherein the anisotropic member comprises a ring and a series of lobes arranged about the ring.

5. The rotating assembly of claim **4** wherein the series of lobes of the anisotropic member impart anisotropic stiffness to the anisotropic member.

6. The rotating assembly of claim **4** wherein the series of lobes comprises lobes positioned at multiple angles about the central axis to generate one or more planes of reflection asymmetry.

7. The rotating assembly of claim **4** wherein the ring of the anisotropic member comprises a split ring.

8. The rotating assembly of claim **4** having an axis perpendicular to the central axis of the through bore of the center housing and wherein the series of lobes of the anisotropic member comprise an alignment with respect to the axis perpendicular to the central axis.

9. The rotating assembly of claim **8** wherein the axis perpendicular to the central axis is aligned with gravity.

10. The rotating assembly of claim **1** wherein the semi-floating bearing comprises a recessed annular region.

11. The rotating assembly of claim **10** wherein the anisotropic member comprises a position at least partially in the recessed annular region.

12. The rotating assembly of claim **10** wherein the semi-floating bearing comprises two recessed annular regions and two of the anisotropic members, each of the anisotropic members positioned at least partially in a respective one of the recessed annular regions.

13. The rotating assembly of claim **10** wherein the semi-floating bearing comprises an opening disposed between two recessed annular regions.

14. The rotating assembly of claim **13** further comprising a locating pin received by the opening disposed between the two recessed annular regions.

15. The rotating assembly of claim **1** further comprising a seal ring positioned in the through bore between the semi-floating bearing and a surface of the center housing wherein the surface defines at least part of the through bore of the center housing.

16. The rotating assembly of claim **1** further comprising multiple seal rings positioned in the through bore between the semi-floating bearing and a surface of the center housing wherein the surface defines at least part of the through bore of the center housing.

17. The rotating assembly of claim **1** wherein the anisotropic member comprises inner spacers and outer spacers.

18. The rotating assembly of claim **1** wherein the anisotropic member comprises a first segment having a first axial width and a second segment having a second axial width and wherein the first axial width differs from the second axial width.

19. A rotating assembly for a turbocharger comprising:

a center housing that comprises a through bore having a central axis;

a semi-floating bearing positioned in the through bore wherein the semi-floating bearing comprises a radial groove and a piston ring seated in the groove that forms a seal between the semi-floating bearing and a surface of the center housing wherein the surface defines at least part of the through bore;

an anisotropic member positioned in the through bore between the semi-floating bearing and a surface of the center housing wherein the surface defines at least part of the through bore of the center housing; and

a shaft, rotatably supported in the center housing by the semi-floating bearing, the shaft connected at one end to a compressor wheel and at another end to a turbine wheel.

20. A rotating assembly for a turbocharger comprising:

a center housing that comprises a through bore having a central axis;

a semi-floating bearing positioned in the through bore;

an anisotropic member positioned in the through bore between the semi-floating bearing and a surface of the center housing wherein the surface defines at least part of the through bore of the center housing and wherein the anisotropic member comprises a ring and a series of lobes arranged about the ring; and

a shaft, rotatably supported in the center housing by the semi-floating bearing, the shaft connected at one end to a compressor wheel and at another end to a turbine wheel.

21. A rotating assembly for a turbocharger comprising:

a center housing that comprises a through bore having a central axis;

a semi-floating bearing positioned in the through bore;

a seal ring and an anisotropic member positioned in the through bore between the semi-floating bearing and a surface of the center housing wherein the surface defines at least part of the through bore of the center housing; and

a shaft, rotatably supported in the center housing by the semi-floating bearing, the shaft connected at one end to a compressor wheel and at another end to a turbine wheel.

22. A rotating assembly for a turbocharger comprising:

a center housing that comprises a through bore having a central axis;

a semi-floating bearing positioned in the through bore;

an anisotropic member positioned in the through bore between the semi-floating bearing and a surface of the center housing wherein the surface defines at least part of the through bore of the center housing wherein the anisotropic member comprises a first segment having a first axial width and a second segment having a second axial width and wherein the first axial width differs from the second axial width; and

a shaft, rotatably supported in the center housing by the semi-floating bearing, the shaft connected at one end to a compressor wheel and at another end to a turbine wheel.

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